VISIT…

# SECTION 22

# MECHANISMS


Machinery Mechanisms ____ 22-2
Slider-Crank Mechanisms ____ 22-11
Torque Factors for Slider Crank Mechanisms ____ 22-19
Mechanisms for Producing Specific Types of Motions ____ 22-20
Traversing Mechanisms Used on Winding Machines ____ 22-22
Mechanisms for Providing Intermittent Rotary Motion ____ 22-24
Intermittent Movements and Mechanisms ____ 22-26
Friction Devices for Intermittent Rotary Motion ____ 22-28
Basic Types of Variable Speed Friction Drives ____ 22-30
Speed Control for Small Mechanisms ____ 22-32
Mechanical Devices for Automatically Governing Speed ____ 22-34
Accelerated and Decelerated Linear Motion Elements ____ 22-36
How to Prevent Reverse Rotation ____ 22-38
Automatic Stopping Mechanisms for Faulty Machine Operation ____ 22-42
Automatic Stop Mechanisms Protect Machine and Work ____ 22-44
Mechanical Automatic Stop Mechanisms ____ 22-46
Automatic Feed Mechanisms for Various Materials ____ 22-48
Automatic Safety Mechanisms for Operating Machines ____ 22-50
Mechanisms Actuated by Air or Hydraulic Cylinders ____ 22-52
Piston-Actuated Mechanisms ____ 22-54
Computing Mechanisms I ____ 22-56
Computing Mechanisms II ____ 22-58
Eight Paper-Feed Mechanisms ____ 22-60
How to Collect and Stack Die Stampings ____ 22-62
Sort, Feed or Weigh Mechanisms ____ 22-64
Spring Motors & Typical Associated Mechanisms ____ 22-68
6 Escapement Mechanisms ____ 22-70
7 more Escapements ____ 22-72
Design Hints for Mechanical Small Mechanisms ____ 22-74

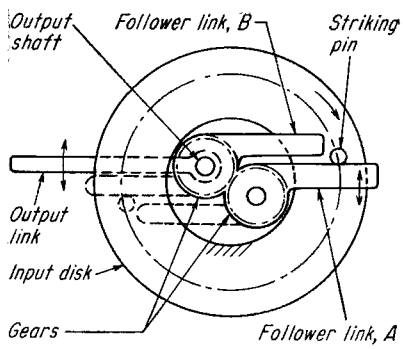
# Machinery Mechanisms

**Selected from industrial and automatic machinery, many of these 41 mechanisms have never appeared in American literature. A diverse grouping—with the very well-known omitted.**

Professor Preben W. Jensen

## REVERSING MECHANISMS

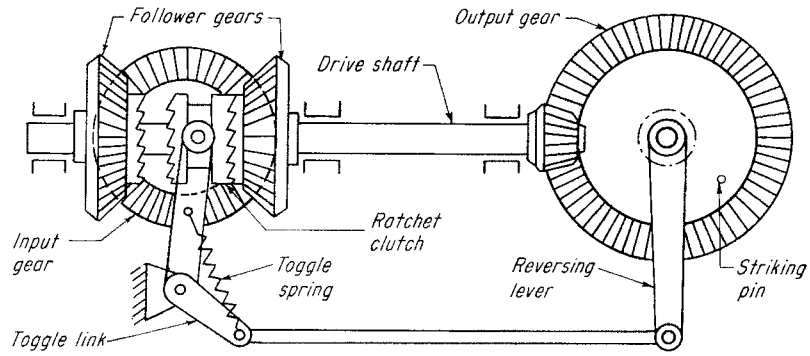
### Double-link reverser




Automatically reverses the output drive every 180-deg rotation of the input. Input disk has press-fit pin which strikes link *A* to drive it clockwise. Link *A* in turn drives link *B* counterclockwise by means of their respective gear segments (or gears pinned to the links). The output shaft and the output link (which may be the working member) are connected to link *B*.

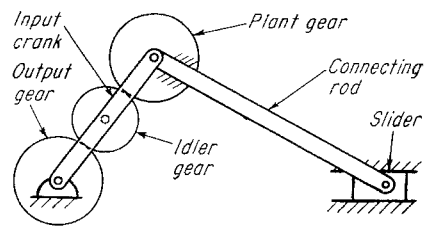
After approximately 180 deg of rotation, the pin slides past link *A* to strike link *B* coming to meet it—and thus reverses the direction of link *B* (and of the output). Then after another 180-deg rotation the pin slips past link *B* to strike link *A* and start the cycle over again.

### Toggle-link reverser




This mechanism also employs a striking pin—but in this case the pin is on the output member. The input bevel gear drives two follower bevels which are free to rotate on their common shaft. The ratchet clutch, however, is spline-connected to the shaft—although free to slide linearly. As shown, it is the right follower gear that is locked to the drive shaft. Hence the output gear rotates clockwise, until the pin strikes the reversing level to shift the toggle to the left. Once past its center, the toggle spring snaps the ratchet to the left to engage the left follower gear. This instantly reverses the output which now rotates counterclockwise until the pin again strikes the reversing level. Thus the mechanism reverses itself every 360-deg rotation of the input.

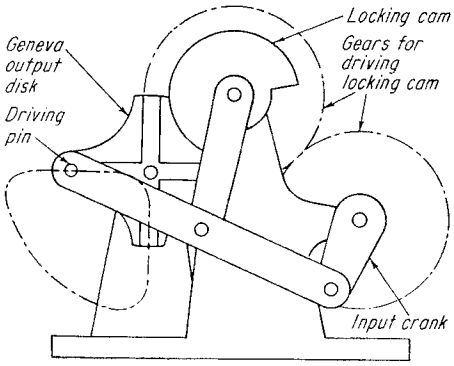
### Modified-Watt's reverser




This is actually a modification of the well-known Watt crank mechanism. The input crank causes the planet gear to revolve around the output gear. But because the planet gear is fixed to the connecting rod it causes the output gear to continually reverse itself. If the radii of the two gears are equal then each full rotation of the input link will cause the output gear to oscillate through same angle as rod.

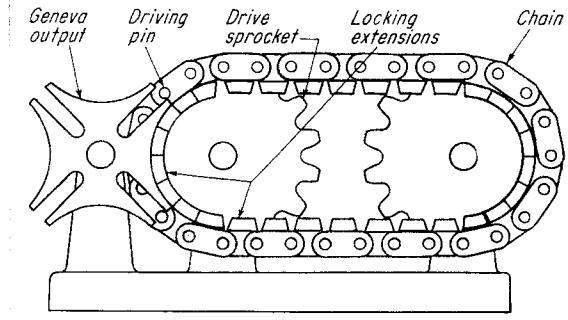
# INTERMITTENT MECHANISMS

**Rapid-transfer geneva**



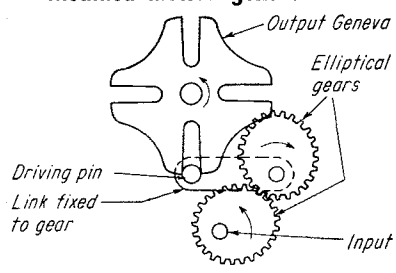

The coupler point at the extension of the connecting link of the 4-bar mechanism describes a curve with two approximately straight lines, 90 deg apart. This provides a good entry situation in that there is no motion in the geneva while the driving pin moves deeply into the slot — then there is an extremely rapid index. A locking cam which prevents the geneva from shifting when it is not indexing, is connected to the input shaft through gears.

**Long-dwell geneva**




This arrangement employs a chain with an extended pin in combination with a standard geneva. This permits a long dwell between each 90-deg shift in the position of the geneva. The spacing between the sprockets determines the length of dwell. Note that some of the links have special extensions to lock the geneva in place between stations.

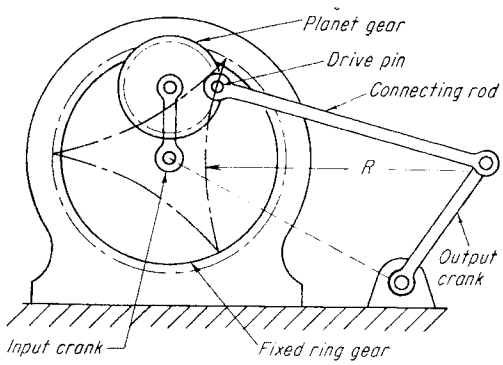
**Modified motion geneva**




With a normal geneva drive the input link rotates at constant velocity, which restricts flexibility in design. That is, for given dimensions and number of stations the dwell period is determined by the speed of the input shaft. Use of elliptical gears produces a varying crank rotation which permits either extending or reducing the dwell period.

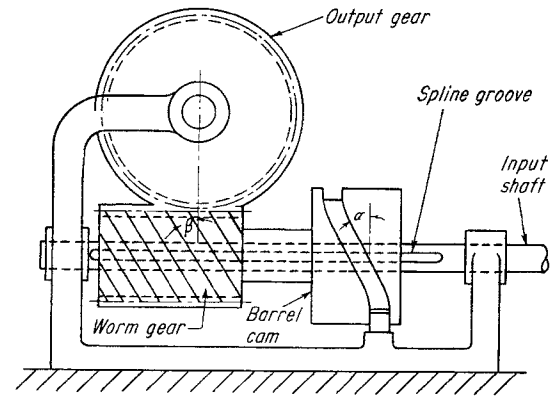
# DWELL MECHANISMS

**Epicyclic dwell mechanism**



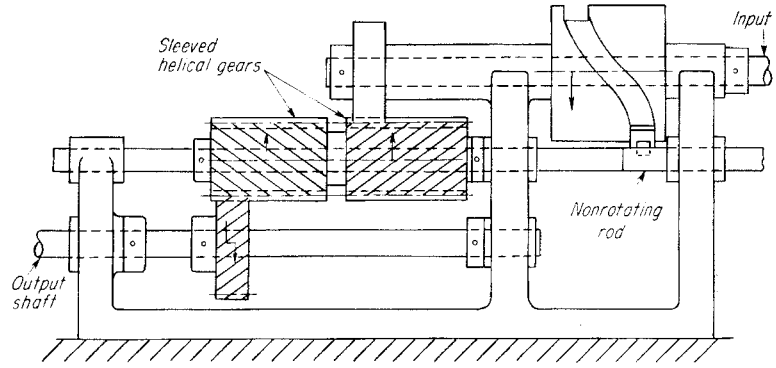

Here the output crank pulsates back and forth with a long dwell at the extreme right position. Input is to the planet gear by means of the rotating crank. The pin on the gear traces the epicyclic three-lobe curve shown. The right portion of the curve is almost a circular arc of radius $R$. If the connecting rod is made equal to $R$, the output crank comes virtually to a standstill during a third of the total rotation of the input. The crank then reverses, comes to a stop at left position, reverses, and repeats dwell.

**Cam-worm dwell mechanism**



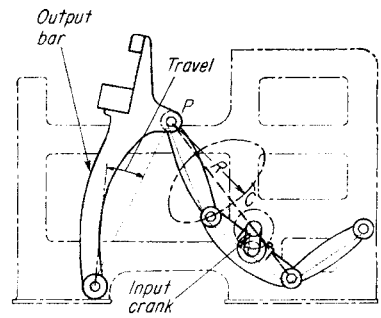

Without the barrel cam, the input shaft would drive the output gear by means of the worm gear at constant speed. The worm and the barrel cam, however, are permitted to slide linearly on the input shaft. Rotation of the input shaft now causes the worm gear to be cammed back and forth, thus adding or subtracting motion to the output. If barrel cam angle $\alpha$ is equal to the worm angle $\beta$, the output comes to a stop during the limits of rotation illustrated, then speeds up to make up for lost time.

### Cam-helical dwell mechanism



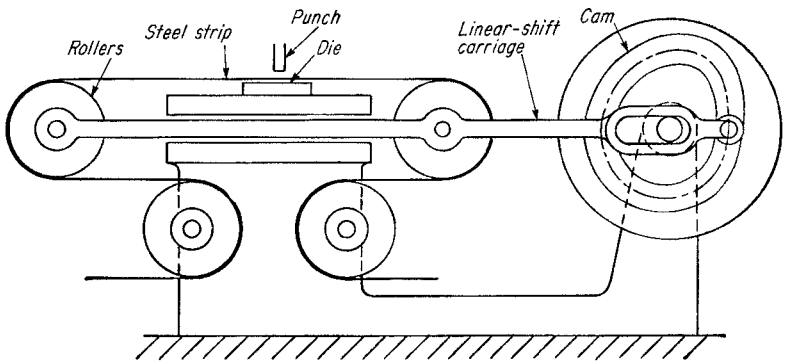

When one helical gear is shifted linearly (but prevented from rotating) it will impart rotary motion to the mating gear because of the helix angle. This principle is used in the mechanism illustrated. Rotation of the input shaft causes the intermediate shaft to shift to the left, which in turn adds or subtracts from the rotation of the output shaft.

### 6-Bar dwell mechanism



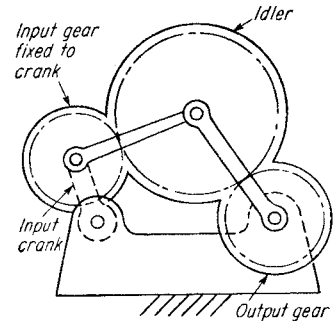

Rotation of the input crank causes the output bar to oscillate with a long dwell at the extreme right position. This occurs because point *C* describes a curve that is approximately a circular arc (from *C* to *C'*) with center at *P*. The output is practically stationary during that portion of curve.

### Cam-roller dwell mechanism



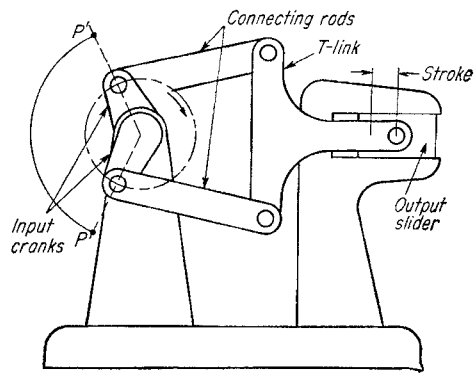

A steel strip is fed at constant linear velocity. But at the die station (illustrated), it is desired to stop the strip so that the punching operation can be performed. The strip passes over movable rollers which, when shifted to the right, cause the strip to move to the right. Since the strip is normally fed to the left, proper design of the cam can nullify the linear feed rates so that the strip stops, and then speeds to catch up to the normal rate.

### Three-gear drive




This is actually a four-bar linkage combined with three gears. As the input crank rotates it takes with it the input gear which drives the output gear by means of the idler. Various output motions are possible. Depending on proportions of the gears, the output gear can pulsate, or come to a short dwell—or even reverse briefly.

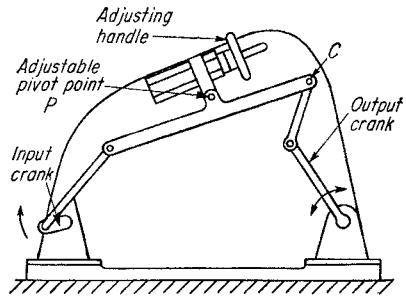
### Double-crank dwell mechanism




Both cranks are connected to a common shaft which also acts as the input shaft. Thus they always remain a constant distance apart from each other. There are only two frame points—the center of the input shaft and the guide for the output slider. As the output slider reaches the end of its stroke (to the right), it remains practically at standstill while one crank rotates through angle *PP'*. Used in textile machinery for cutting.

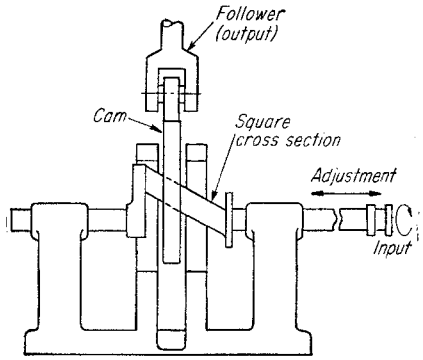
# ADJUSTABLE-OUTPUT MECHANISMS

### Linkage motion adjuster



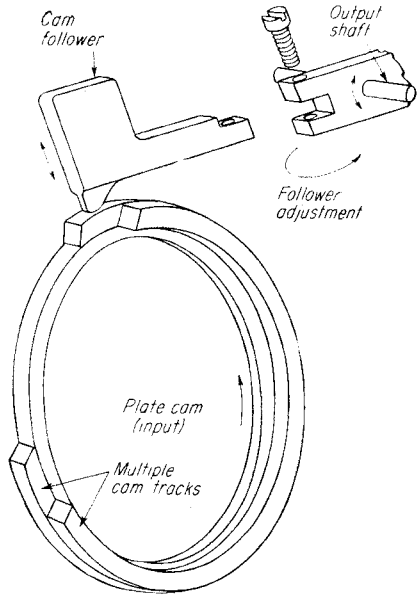

Here the motion and timing of the output link can be varied *during operation* by shifting the pivot point of the intermediate link of the 6-bar linkage illustrated. Rotation of the input crank causes point *C* to oscillate around the pivot point *P*. This in turn imparts an oscillating motion to the output crank. Shifting of point *P* is accomplished by a screw device.

### Cam motion adjuster



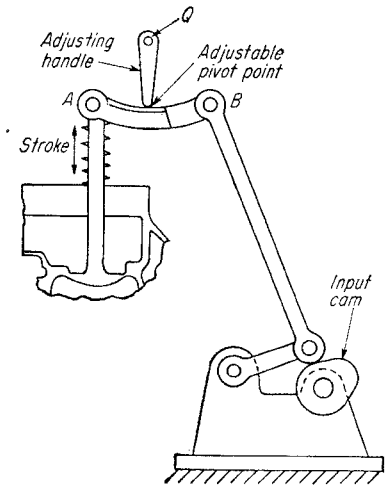

The output motion of the cam follower is varied by linearly shifting the input shaft to the right or left during operation. The cam has a square hole which fits over the square cross section of the crank shaft. Rotation of the input shaft causes eccentric motion in the cam. Shifting the input shaft to the right, for instance, causes the cam to move radially outward, thus increasing the stroke of the follower.

### Double-cam mechanism



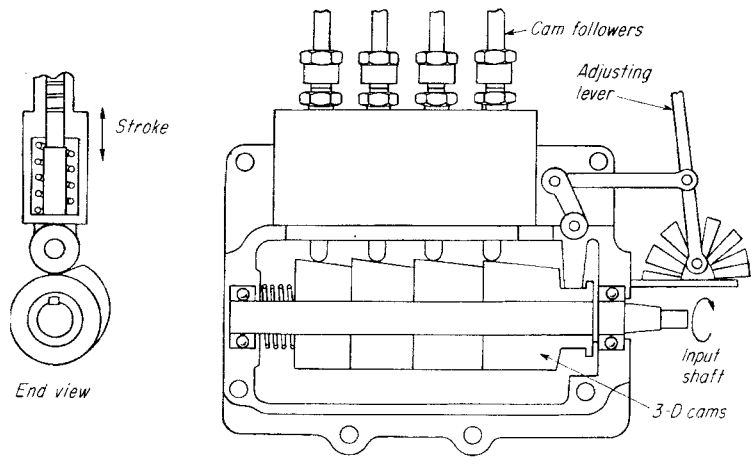

This is a simple but effective technique for changing the timing of a cam. The follower can be adjusted in the horizontal plane but is restricted in the vertical plane. The plate cam contains two or more cam tracks.

### Valve stroke adjuster



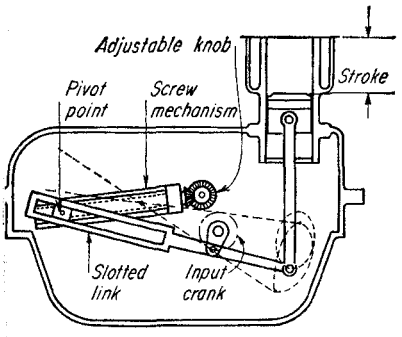

This mechanism adjusts the stroke of valves of combustion-engines. One link has a curved surface and pivots around an adjustable pivot point. Rotating the adjusting link changes the proportion of strokes of points *A* and *B* and hence of the valve. The center of curvature of the curve link is at point *Q*.

### 3-D mechanism



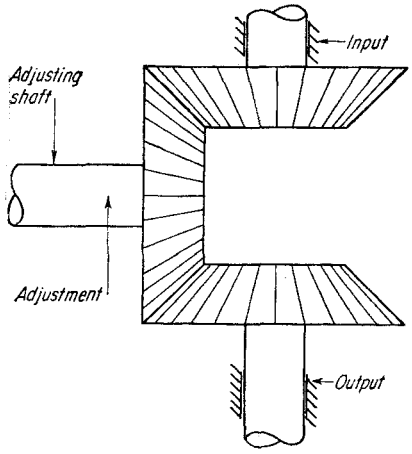

Output motions of four followers are varied during the rotation by shifting the quadruple 3-D cam to the right or left. Linear shift is made by means of adjustment lever which can be released in any of the 6 positions.

**Piston-stroke adjuster**



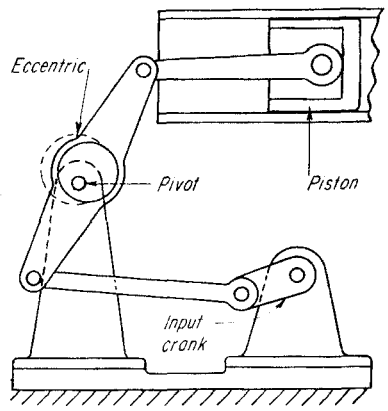

The input crank oscillates the slotted link to drive the piston up and down. The position of the pivot point can be adjusted by means of the screw mechanism even while the piston is under full load.

**Shaft synchronizer**




Actual position of the adjusting shaft is normally kept constant. The input then drives the output by means of the bevel gears. Rotating the adjusting shaft in a plane at right angle to the input-output line changes the relative radial position of the input and output shafts, used for introducing a torque into the system while running, synchronizing the input and output shafts, or changing the timing of a cam on the output shaft.

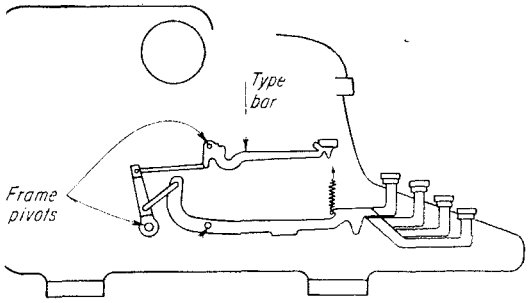
**Eccentric pivot point**




Rotation of the input crank reciprocates the piston. The stroke depends on position of the pivot point which is easily adjusted, even during rotation, by rotating the eccentric shaft.

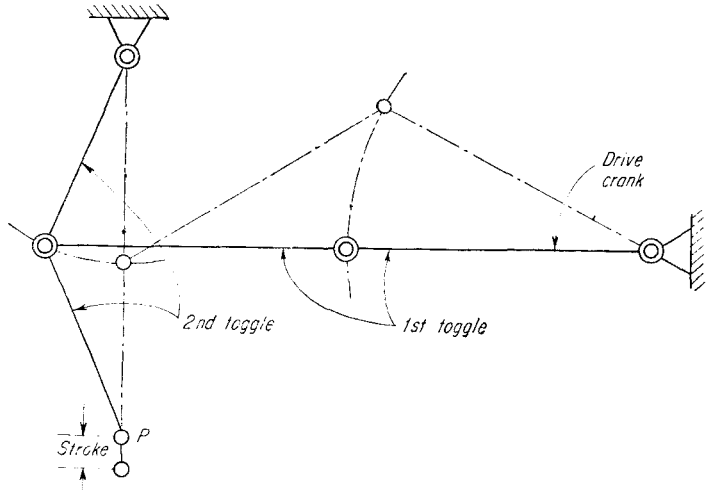
# FORCE AND STROKE MULTIPLIERS

**Typewriter drive**



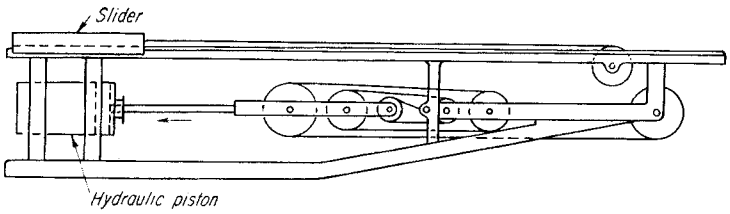

Multiplies the finger force in a typewriter, thus producing a strong hammer action at the roller from a light touch. There are three pivot points attached to the frame. Arrangement of the links is such that the type bar can move in free flight after a key has been struck. The mechanism illustrated is actually two four-bar linkages in series. In certain typewriters, as many as four 4-bar linkages are used in a series.

**Double-toggle puncher**




The first toggle keeps point *P* in the raised position even though its weight may be exerting a strong downward force (as in a heavy punch weight). When the drive crank rotates clockwise (driven, say, by a reciprocating mechanism), the second toggle begins to straighter to create a strong punching force.

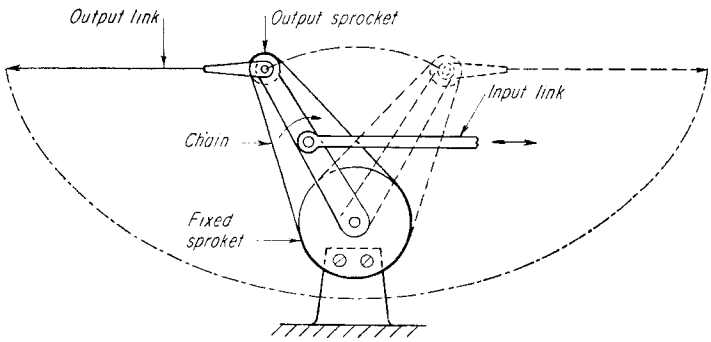
### Pulley drive




A simple arrangement which multiplies the stroke of a hydraulic piston, causing the slider to propel rapidly to the right for catapulting.

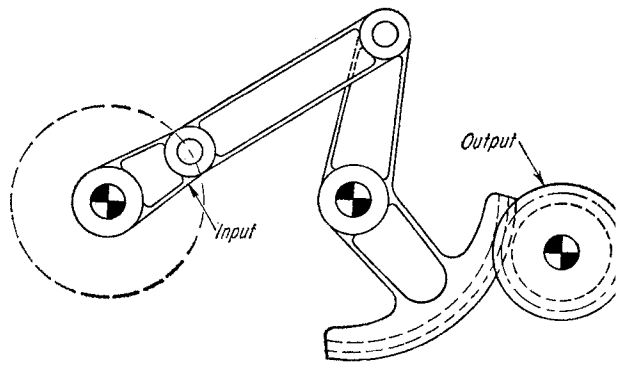
## ANGLE MULTIPLIERS

### Wide angle oscillator



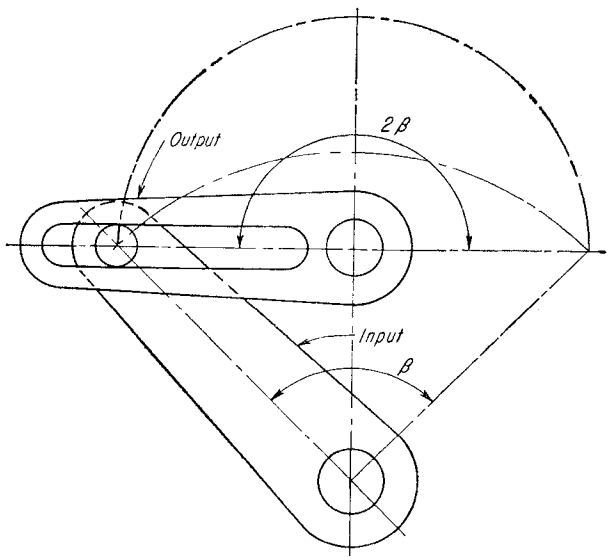

Motion of the input linkage is converted into a wide angle oscillation by means of the two sprockets and chain in the diagram illustrated. An oscillation of 60 deg is converted into 180-deg oscillation.

### Gear-sector drive



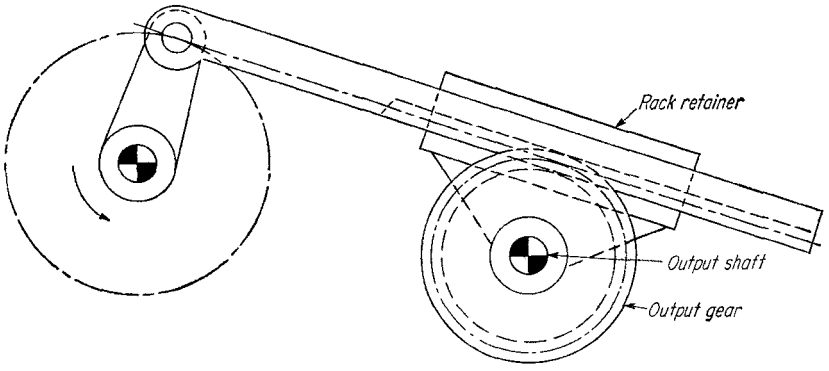

This is actually a four-bar linkage combined with a set of gears. A four-bar linkage usually can get no more than about 120-deg maximum oscillation. The gear segments multiply the oscillation in inverse proportion to the radii of the gears. For the proportions shown, the oscillation is boosted 2½ times.

### Angle-doubling drive



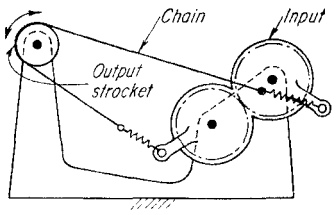

Frequently it is desired to enlarge the oscillating motion $\beta$ of one machine member into an output oscillation of, say, $2\beta$. If gears are employed, the direction of rotation cannot be the same unless an idler gear is used, in which case the centers of the intput and output shafts cannot be too close. Rotating the input link clockwise causes the output to also follow in a clockwise direction. For a particular set of link proportions, distance between the shafts determines the gain in angle multiplication.

### Gear-rack drive



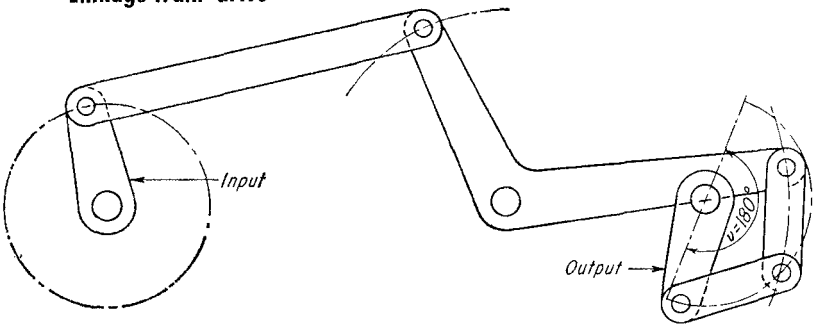

This mechanism is frequently employed to convert the motion of an input crank into a much larger rotation of the output (say, 30 to 360 deg). The crank drives the slider and gear rack, which in turn rotates the output gear.

### Chain drive




Springs and chains are attached to geared cranks to operate a sprocket output. Depending on the gear ratio, the output will produce a specified oscillation, say two revolutions of output in each direction for each 360 deg of input.

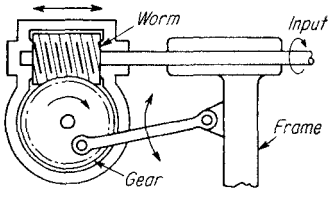
### Linkage-train drive




Arranging linkages in series can increase the angle of oscillation. In the case illustrated the oscillating motion of the L-shaped rocker is the input for the second linkage. Final oscillation is 180 deg.

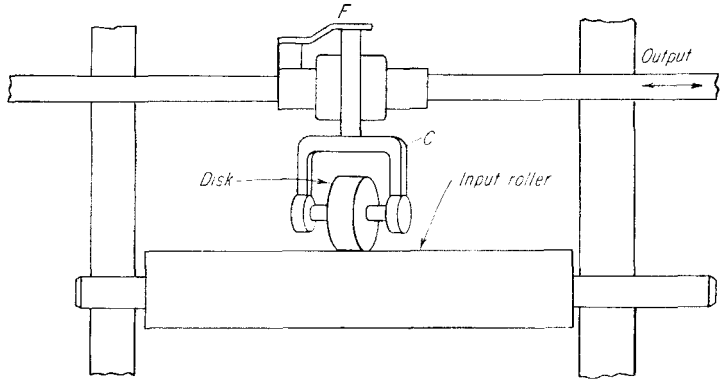
## ROTARY TO LINEAR-MOTION MECHANISMS

### Linear reciprocator



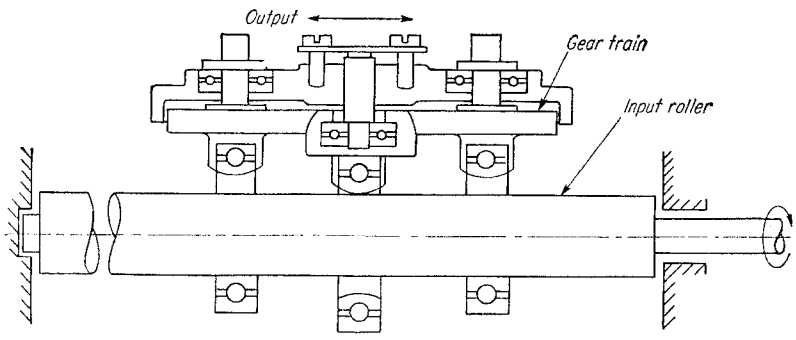

The objective here is to convert a rotary motion into a reciprocating motion that is *in line* with the input shaft. Rotation of the shaft drives the worm gear which is attached to the machine frame by means of a rod. Thus input rotation causes the worm gear to draw itself (and the worm) to the right—thus providing a back and forth motion. Employed in connection with a color-transfer cylinder in printing machines.

### Disk and roller drive



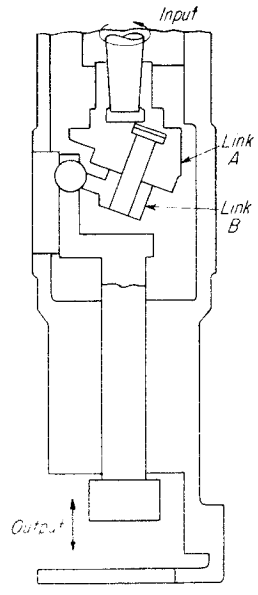

Here a hardened disk, riding at an angle to an input roller, transforms the rotary motion into linear motion *parallel to the axis of the input*. The roller is pressed against the input shaft with the help of a flat spring, *F*. Feed rate is easily varied by changing the angle of the disk. Arrangement can produce an extremely slow feed with a built-in safety factor in case of possible jamming.

### Bearings and roller drive



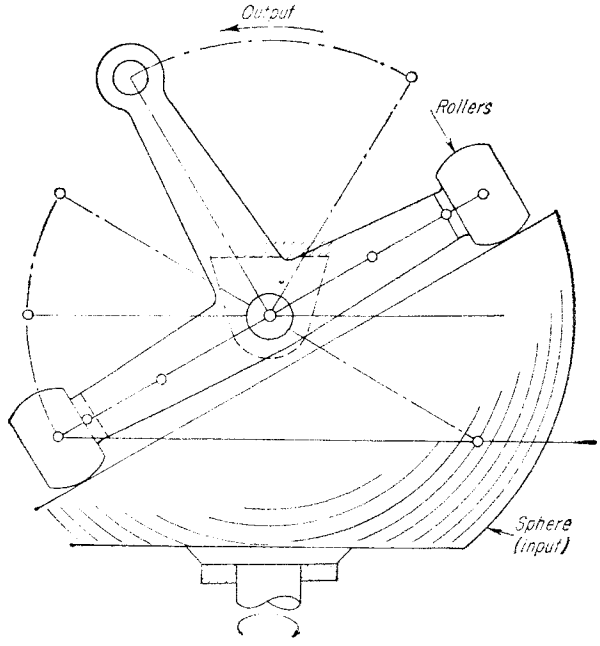

Similar to the previous one, this arrangement avoids large Hertzian stresses between disk and roller by employing three ball bearings in place of the single disk. The inner races of the bearings make contact on one side or the other. Hence a gearing arrangement is required to alternate the angle of the bearings. This arrangement also reduces the bending moment on the shaft.

### Reciprocating space crank



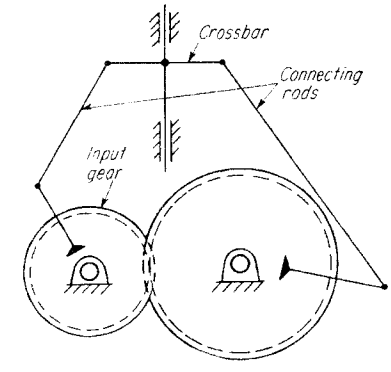

Rotary input causes the bottom surface of link *A* to wobble in reference to the center line. Link *B* is free of link *A* but restrained from rotating by the slot. This causes the output member to reciprocate linearly. Employed for filing machine.

### Sinusoidal reciprocator



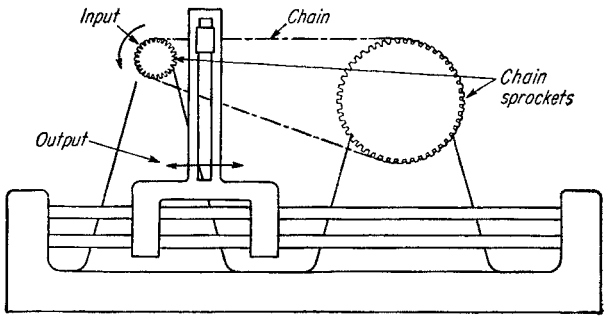

This transforms rotary motion into a reciprocating motion in which the oscillating output member is in the same plane as the input shaft. The output member has two arms with rollers which contact the surface of the truncated sphere. Rotation of the sphere causes the output to oscillate.

### Cross-bar reciprocator




Although complex-looking, this device has been successful in high speed machines for transforming rotary motion into a high-impact linear motion. Both gears contain cranks connected to the cross bar by means of connecting rods.

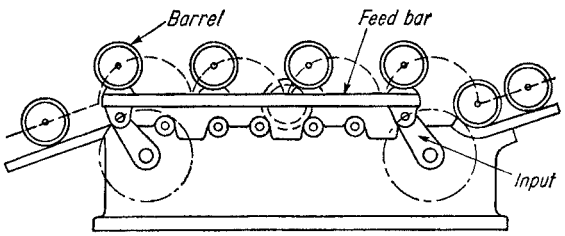
**Two-speed reciprocator**




A drive pin, fastened to a chain, rides in a vertical slot to reciprocate a carriage in a horizontal direction. Velocity in both directions is constant and depends on the angle of slope that the chain makes with the vertical slot. Thus, velocity to the left is higher than to the right because the top of the chain is horizontal.

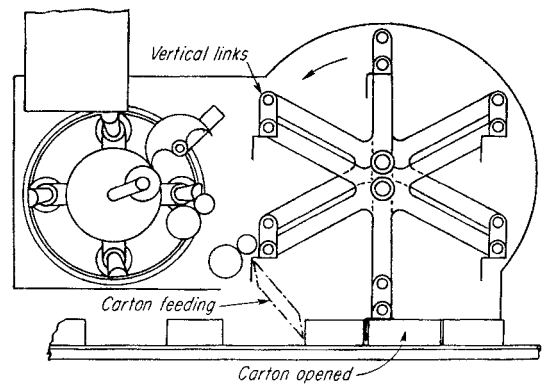
# PARALLEL LINK MECHANISMS

**Parallel link feeder**



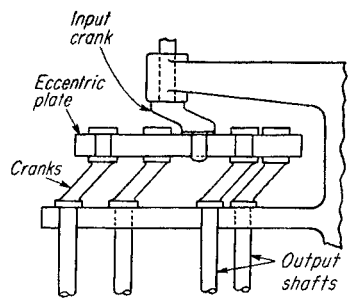

One of the cranks is the input, the other follows to keep the feeding bar horizontal. Employed for moving barrels from station to station.

**Parallelogram linkage**



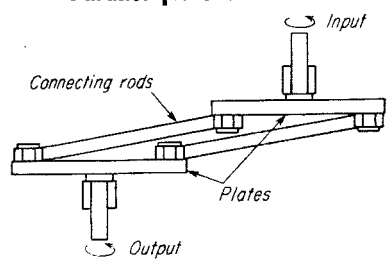

All seven short links are kept in a vertical position while rotating. The center link is the driver. This particular application feeds and opens cartons, but the device is capable of many diverse applications.

**Parallel link driller**




For powering a group of shafts. The input crank drives the eccentric plate. This in turn rotates the output cranks, all of which are of the same length and rotate at the same speed. Gears would require more room between shafts and are usually more expensive.

**Parallel plate driver**




Here again, the input and output rotate with the same angle of relationship. The position of the shafts, however, can vary to suit other requirements without affecting the input-output relationship between shafts.

# Slider-Crank Mechanisms

## New equations and computer-tabulated values of velocities, accelerations, and forces reduce computation time an errors.

Professor Merl D. Creech

THE slider crank—an efficient mechanism for changing reciprocating motion to rotary—is widely used in engines, pumps, automatic machinery, and machine tools. The crank is a special case of the basic four-bar kinematic chain in which one of the four bars is of infinite length, ie, the motion of a point moving along a straight line is equivalent to the rotation of the same point about a center at infinity.

To design a slider-crank mechanism for high-speed operation, it is necessary to calculate the varying angular velocities and accelerations of the connecting rod and the linear accelerations of the slider to determine the maximum inertia forces acting on the system. Such time-consuming calculations frequently lead to errors.

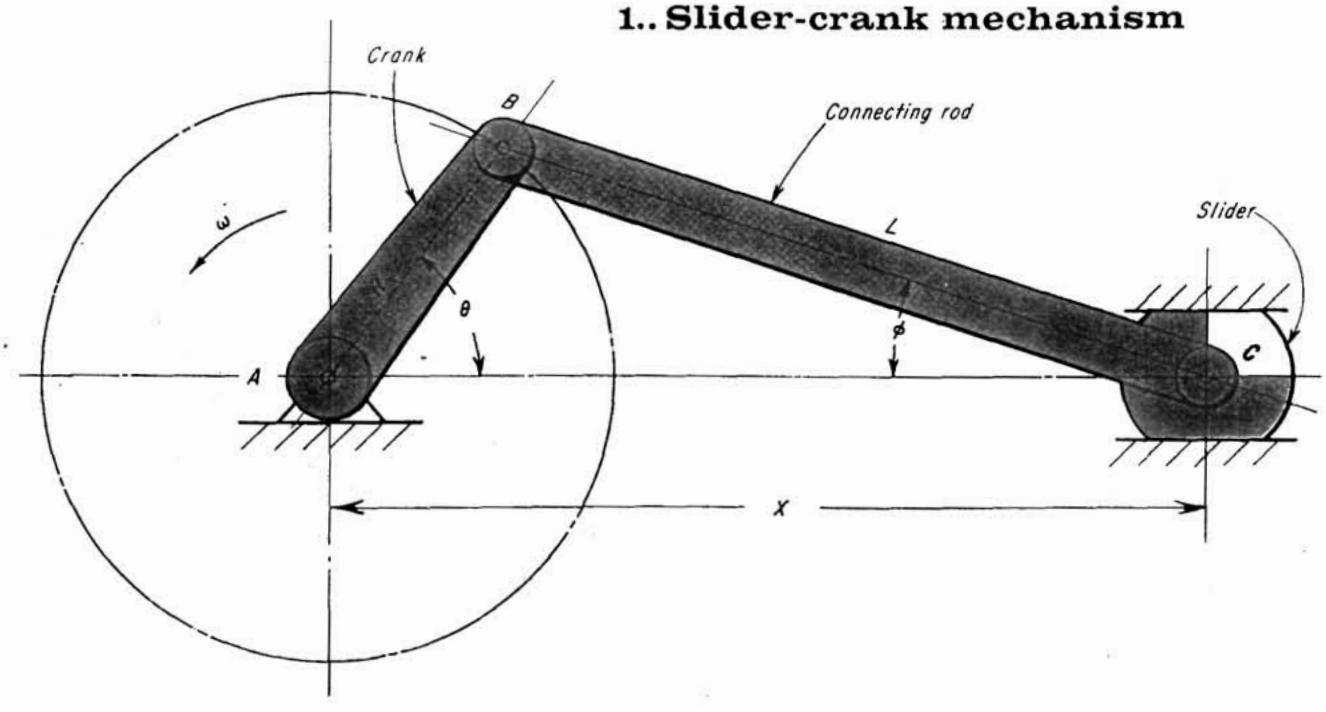
The equations developed here for finding such factors are in a more streamlined form than is generally available. They have been programmed on a computer to obtain a set of design tables. Tabulations are carried to four significant figures and accurate interpolation is possible when required. Two numerical examples illustrate the use of the tables.

### Derivation of equations

The effect of the angular acceleration of the crankshaft on the angular velocities and accelerations of the connecting rod and the acceleration of the slider is quite small. Hence it is assumed to be zero in the derivation.

From the geometry of the system in Fig 1 (see also

**1.. Slider-crank mechanism**



list of symbols), the relationship between $\theta$ and $\phi$ is

$$R \sin \theta = L \sin \phi \quad (1)$$

and

$$x = R \cos \theta + L \cos \phi \quad (2)$$

Differentiating Eq 1 with respect to time gives

$$\theta'(R/L) \cos \theta = \phi' \cos \phi \quad (3)$$

Also from Eq 1 we can write

$$(R/L)^2 \sin^2 \theta = \sin^2 \phi = 1 - \cos^2 \phi \quad (4)$$

Solving for $\cos \phi$ gives

$$\cos \phi = \left[1 - \left(\frac{R}{L}\right)^2 \sin^2 \theta\right]^{1/2} \quad (5)$$

Since $\omega = \theta'$, Eq 3 and 5 can be combined to give

**Angular velocity of the connecting rod**

$$\phi' = \omega \left[\frac{(R/L) \cos \theta}{[1 - (R/L)^2 \sin^2 \theta]^{1/2}}\right] \quad (6)$$

Now, differentiating Eq 2 with respect to time

$$\frac{x'}{L} = -\theta' \left(\frac{R}{L}\right) \sin \theta - \phi' \sin \phi \quad (7)$$

Substituting Eq 1 into Eq 7 gives

**Linear velocity of the piston**

$$\frac{x'}{L} = -\omega \left[1 + \frac{\phi'}{\omega}\right] \left(\frac{R}{L}\right) \sin \theta \quad (8)$$

Differentiating Eq 3 with respect to time

$$-(\theta')^2 \left(\frac{R}{L}\right) \sin \theta + \theta'' \left(\frac{R}{L}\right) \cos \theta = \phi'' \cos \phi - (\phi')^2 \sin \phi \quad (9)$$

Since $\omega = \theta' =$ constant and $\theta'' = 0$, the values of $\cos \phi$ from Eq 5, $\sin \theta$ from Eq 1, and $\phi'$ from Eq 6 are substituted into Eq 9 to give

**Angular acceleration of the connecting rod**

$$\phi'' = \frac{\omega^2 (R/L) \sin \theta [(R/L)^2 - 1]}{[1 - (R/L)^2 \sin^2 \theta]^{3/2}} \quad (10)$$

Next, differentiating Eq 7 with respect to time

$$\frac{x''}{L} = -\theta'' \left(\frac{R}{L}\right) \sin \theta - (\theta')^2 \left(\frac{R}{L}\right) \cos \theta - \phi'' \sin \phi - (\phi')^2 \cos \phi \quad (11)$$

Substituting the value of $\phi'$ from Eq 6, $\sin \phi$ from Eq 4, $\cos \phi$ from Eq 5 and $\phi''$ from Eq 10 into Eq 11 gives

**Slider acceleration**

$$\frac{x''}{L} = -\omega^2 \left(\frac{R}{L}\right) \left[\cos \theta + \frac{\phi''}{\omega^2} \sin \theta + \frac{\phi'}{\omega} \cos \theta\right] \quad (12)$$

**Dynamic forces**

The resultant force, $F_R$, acting on the connecting rod, Fig 2, is due to the angular and linear accelerations, $\phi''$ and $x''$ respectively. The effect of gravity on the connecting rod is small in comparison to these dynamic forces and can be neglected. Thus, resultant force (see Symbols in Fig 3) is equal to

$$F_R = Mr\phi'' + Mx'' \sin \phi \quad (13)$$

The moments acting on the connecting rod are

$$F_R d = MK^2 \phi'' + Mrx'' \sin \phi \quad (14)$$

where

$$K^2 = I_C / M \quad (15)$$

Hence the distance of the resultant, $F_R$, from point $C$ is equal to

$$d = \frac{K^2 \phi'' + rx'' \sin \phi}{r\phi'' + x'' \sin \phi} \quad (16)$$

The signs of the terms in the above equations are correct for counterclockwise rotation of the crank shaft and for values of $\theta$ in the first quadrant.

There are also dynamic forces in the longitudinal direction (along the line of centers, $BC$). The resultant longitudinal force, $F_L$, can be written as

$$F_L = Mr(\phi')^2 - Mx'' \cos \phi \quad (17)$$

**Tabulated values**

From considerations of symmetry it is necessary only to tabulate the values of the velocities and accelerations for values of $\theta$ from 0 deg to 180 deg. But there is always the problem of signs—to avoid error, check with Fig 3, which shows the directions of the acceleration and velocity of the connecting rod and slider for clockwise and counterclockwise rotation of the crank.

Table I gives the relationship between the design equations and the values tabulated in Tables II to VII. These values, grouped under $R/L$ ratios of 0.2, 0.3, 0.4, 0.5, 0.6 and 0.7, will handle most applications that may arise and the designer can always interpolate between values if necessary.

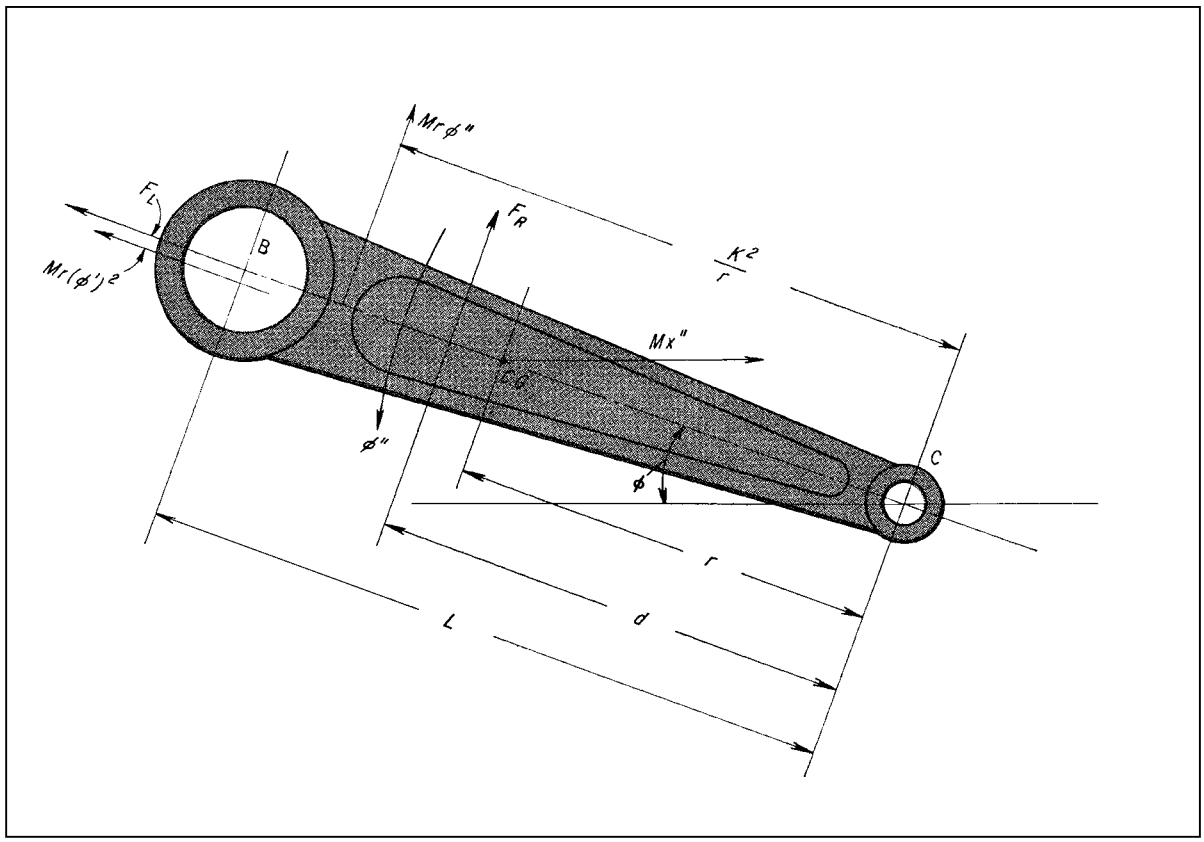
**Example I—reciprocating engine**

Stroke of engine = 7.2 in.
Connecting-rod length = 12.0 in.

SYMBOLS

$L$ = length of connecting rod
$R$ = crank length; radius of crank circle
$x$ = distance from center of crankshaft, $A$, to wrist pin, $C$
$x'$ = slider velocity (linear velocity of point $C$)
$x''$ = slider acceleration
$\theta$ = crank angle measured from dead center (when slider is fully extended)
$\theta'$ = crank angular velocity = $d\theta/dt$
$\theta''$ = crank angular acceleration = $d^2\theta/dt^2$
$\phi$ = angular position of connecting rod; $\phi = 0$ when $\theta = 0$
$\phi'$ = connecting-rod angular velocity = $d\phi/dt$
$\phi''$ = connecting-rod angular acceleration = $d^2\phi/dt^2$
$\omega$ = constant crank angle velocity

## 2.. Dynamic forces on connecting rod




Connecting-rod weight = 8.106 lb.

Center of mass of connecting rod = 6.4 in. from center of wrist pin.

Moment of inertia about the center of wrist pin = 1.1088 in.-lb-sec².

**Part A:** Find the direction, magnitude, and location of the inertia forces acting on the connecting rod when the crank is 35 deg from top dead center and the engine speed is 2400 rpm.

$$R/L = 3.6/12 = 0.30$$

$$\omega = \frac{(2400)(2\pi)}{60} = 251.33 \text{ rad/sec}$$

The following values of the required parameters are found from Table III for $\theta = 35$ deg:

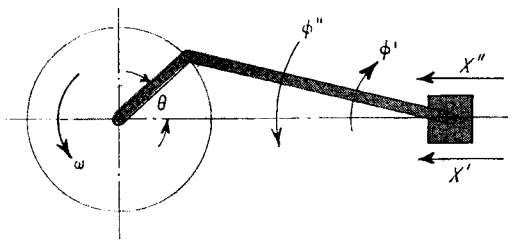
$$\phi = 9.908 \text{ deg}, \quad \phi'/\omega = 0.2495, \quad \phi''/\omega^2 L = -0.1638,$$
$$x'/\omega L = -0.2150, \quad x''/\omega^2 L = 0.2789$$

Using these tabulated values, the following terms are calculated:

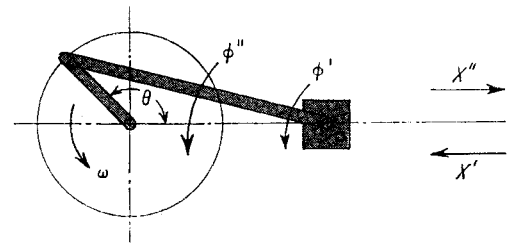
TABLE **I.. Relation between equations and tabulated values**

| CONNECTING ROD | | | | SLIDER | | | |
|---|---|---|---|---|---|---|---|
| | SYMBOL | EQ. NO. | COLUMN | | SYMBOL | EQ. NO. | COLUMN |
| ANGULAR DISPLACEMENT | $\phi$ | 5 | 2 | LINEAR DISPLACEMENT | $x$ | 2 | — |
| ANGULAR VELOCITY | $\phi'/\omega$ | 6 | 3 | LINEAR VELOCITY | $x'/\omega L$ | 8 | 4 |
| ANGULAR ACCELERATION | $\phi''/\omega^2$ | 10 | 5 | LINEAR ACCELERATION | $x''/\omega^2 L$ | 12 | 6 |

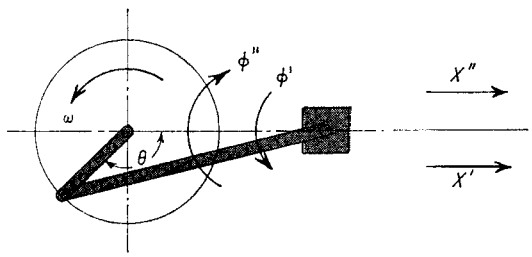
## 3.. Directions of velocities and accelerations



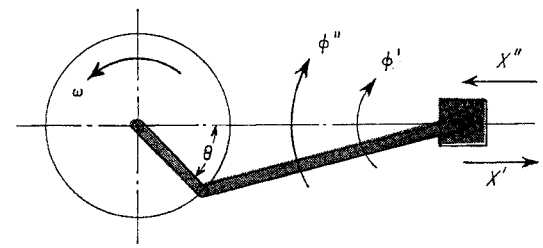

(A) $0 < \theta < 90°$




(B) $90° < \theta < 180°$




(C) $180° < \theta < 270°$



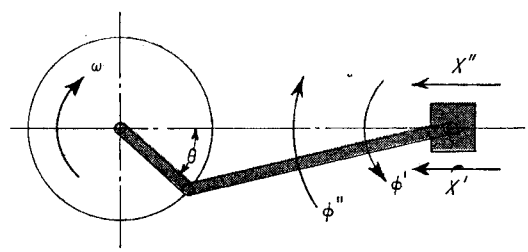

(D) $270° < \theta < 360°$

Counterclockwise rotation

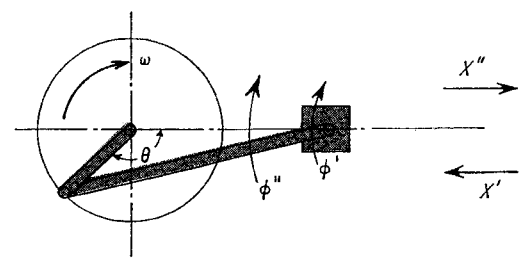
Clockwise rotation

SYMBOLS

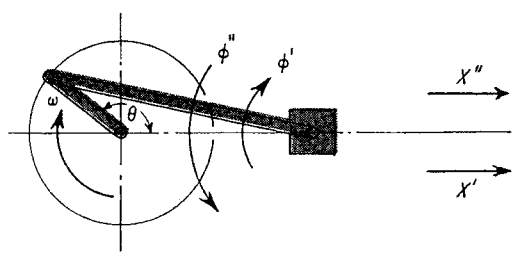
$d$ = distance of resultant $F_R$ from $C$
$F_R$ = resultant of forces
$I_C$ = moment of inertia of connecting rod about $C$
$K$ = radius of gyration from $C$
$M$ = mass of connecting rod
$r$ = distance of center of gravity from $C$



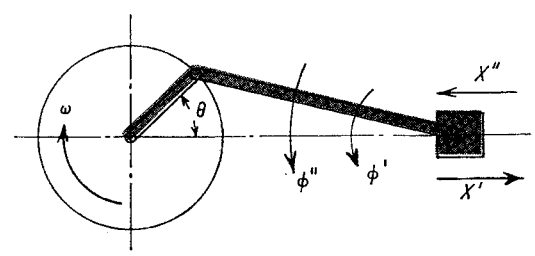

(A) $270° < \theta < 360°$




(B) $180° < \theta < 270°$




(C) $90° < \theta < 180°$




(D) $0° < \theta < 90°$

TABLE II
$R/L=0.2$

| $\theta$, deg | $\phi$, deg | $\phi'/\omega$ | $\chi'/\omega L$ | $\phi''/\omega^2$ | $\chi''/\omega^2 L$ |
|---|---|---|---|---|---|
| 0 | 0.000 | 0.2000 | −0.0000 | 0.0000 | 0.2400 |
| 5 | 0.998 | 0.1993 | −0.0209 | −0.0167 | 0.2386 |
| 10 | 1.990 | 0.1971 | −0.0416 | −0.0334 | 0.2346 |
| 15 | 2.967 | 0.1934 | −0.0618 | −0.0499 | 0.2280 |
| 20 | 3.922 | 0.1884 | −0.0813 | −0.0661 | 0.2188 |
| 25 | 4.848 | 0.1819 | −0.0999 | −0.0820 | 0.2073 |
| 30 | 5.739 | 0.1741 | −0.1174 | −0.0975 | 0.1936 |
| 35 | 6.587 | 0.1649 | −0.1336 | −0.1123 | 0.1780 |
| 40 | 7.386 | 0.1545 | −0.1484 | −0.1265 | 0.1606 |
| 45 | 8.130 | 0.1429 | −0.1616 | −0.1399 | 0.1418 |
| 50 | 8.812 | 0.1301 | −0.1731 | −0.1524 | 0.1219 |
| 55 | 9.429 | 0.1163 | −0.1829 | −0.1638 | 0.1012 |
| 60 | 9.974 | 0.1015 | −0.1908 | −0.1740 | 0.0800 |
| 65 | 10.443 | 0.0859 | −0.1968 | −0.1830 | 0.0586 |
| 70 | 10.832 | 0.0696 | −0.2010 | −0.1904 | 0.0374 |
| 75 | 11.138 | 0.0528 | −0.2034 | −0.1963 | 0.0166 |
| 80 | 11.359 | 0.0354 | −0.2039 | −0.2006 | −0.0036 |
| 85 | 11.492 | 0.0178 | −0.2028 | −0.2032 | −0.0228 |
| 90 | 11.536 | 0.0000 | −0.2000 | −0.2041 | −0.0408 |
| 95 | 11.492 | −0.0178 | −0.1957 | −0.2032 | −0.0576 |
| 100 | 11.359 | −0.0354 | −0.1900 | −0.2006 | −0.0730 |
| 105 | 11.138 | −0.0528 | −0.1830 | −0.1963 | −0.0870 |
| 110 | 10.832 | −0.0696 | −0.1748 | −0.1904 | −0.0994 |
| 115 | 10.443 | −0.0859 | −0.1657 | −0.1830 | −0.1104 |
| 120 | 9.974 | −0.1015 | −0.1556 | −0.1740 | −0.1200 |
| 125 | 9.429 | −0.1163 | −0.1448 | −0.1638 | −0.1282 |
| 130 | 8.812 | −0.1301 | −0.1333 | −0.1524 | −0.1352 |
| 135 | 8.130 | −0.1429 | −0.1212 | −0.1399 | −0.1410 |
| 140 | 7.386 | −0.1545 | −0.1087 | −0.1265 | −0.1458 |
| 145 | 6.587 | −0.1649 | −0.0958 | −0.1123 | −0.1497 |
| 150 | 5.739 | −0.1741 | −0.0826 | −0.0975 | −0.1528 |
| 155 | 4.848 | −0.1819 | −0.0691 | −0.0820 | −0.1552 |
| 160 | 3.922 | −0.1884 | −0.0555 | −0.0661 | −0.1571 |
| 165 | 2.967 | −0.1934 | −0.0417 | −0.0499 | −0.1584 |
| 170 | 1.990 | −0.1971 | −0.0279 | −0.0334 | −0.1593 |
| 175 | 0.998 | −0.1993 | −0.0140 | −0.0167 | −0.1598 |
| 180 | 0.000 | −0.2000 | −0.0000 | −0.0000 | −0.1600 |

TABLE III
$R/L=0.3$

| $\theta$, deg | $\phi$, deg | $\phi'/\omega$ | $\chi'/\omega L$ | $\phi''/\omega^2$ | $\chi''/\omega^2 L$ |
|---|---|---|---|---|---|
| 0 | 0.000 | 0.3000 | −0.0000 | 0.0000 | 0.3900 |
| 5 | 1.498 | 0.2990 | −0.0340 | −0.0238 | 0.3876 |
| 10 | 2.986 | 0.2958 | −0.0675 | −0.0476 | 0.3804 |
| 15 | 4.453 | 0.2907 | −0.1002 | −0.0713 | 0.3685 |
| 20 | 5.889 | 0.2834 | −0.1317 | −0.0949 | 0.3520 |
| 25 | 7.283 | 0.2741 | −0.1615 | −0.1182 | 0.3314 |
| 30 | 8.626 | 0.2628 | −0.1894 | −0.1412 | 0.3069 |
| 35 | 9.908 | 0.2495 | −0.2150 | −0.1638 | 0.2789 |
| 40 | 11.118 | 0.2342 | −0.2380 | −0.1857 | 0.2478 |
| 45 | 12.247 | 0.2171 | −0.2581 | −0.2068 | 0.2143 |
| 50 | 13.286 | 0.1981 | −0.2753 | −0.2269 | 0.1789 |
| 55 | 14.225 | 0.1775 | −0.2894 | −0.2455 | 0.1423 |
| 60 | 15.058 | 0.1553 | −0.3002 | −0.2626 | 0.1051 |
| 65 | 15.776 | 0.1317 | −0.3077 | −0.2776 | 0.0680 |
| 70 | 16.374 | 0.1069 | −0.3120 | −0.2905 | 0.0317 |
| 75 | 16.844 | 0.0811 | −0.3133 | −0.3008 | −0.0032 |
| 80 | 17.184 | 0.0545 | −0.3116 | −0.3083 | −0.0361 |
| 85 | 17.389 | 0.0274 | −0.3070 | −0.3129 | −0.0667 |
| 90 | 17.457 | 0.0000 | −0.3000 | −0.3145 | −0.0943 |
| 95 | 17.389 | −0.0274 | −0.2907 | −0.3129 | −0.1189 |
| 100 | 17.184 | −0.0545 | −0.2793 | −0.3083 | −0.1403 |
| 105 | 16.844 | −0.0811 | −0.2663 | −0.3008 | −0.1585 |
| 110 | 16.374 | −0.1069 | −0.2518 | −0.2905 | −0.1735 |
| 115 | 15.776 | −0.1317 | −0.2361 | −0.2776 | −0.1856 |
| 120 | 15.058 | −0.1553 | −0.2194 | −0.2626 | −0.1949 |
| 125 | 14.225 | −0.1775 | −0.2021 | −0.2455 | −0.2019 |
| 130 | 13.286 | −0.1981 | −0.1843 | −0.2269 | −0.2069 |
| 135 | 12.247 | −0.2171 | −0.1661 | −0.2068 | −0.2099 |
| 140 | 11.118 | −0.2342 | −0.1477 | −0.1857 | −0.2118 |
| 145 | 9.908 | −0.2495 | −0.1291 | −0.1638 | −0.2126 |
| 150 | 8.626 | −0.2628 | −0.1106 | −0.1412 | −0.2127 |
| 155 | 7.283 | −0.2741 | −0.0920 | −0.1182 | −0.2124 |
| 160 | 5.889 | −0.2834 | −0.0735 | −0.0949 | −0.2117 |
| 165 | 4.453 | −0.2907 | −0.0551 | −0.0713 | −0.2111 |
| 170 | 2.986 | −0.2958 | −0.0367 | −0.0476 | −0.2105 |
| 175 | 1.493 | −0.2990 | −0.0183 | −0.0238 | −0.2101 |
| 180 | 0.000 | −0.3000 | −0.0000 | −0.0000 | −0.2100 |

TABLE IV
$R/L=0.4$

| $\theta$, deg | $\phi$, deg | $\phi'/\omega$ |
|---|---|---|
| 0 | 0.000 | 0.4000 |
| 5 | 1.997 | 0.0987 |
| 10 | 3.982 | 0.3949 |
| 15 | 5.942 | 0.3885 |
| 20 | 7.863 | 0.3794 |
| 25 | 9.732 | 0.3678 |
| 30 | 11.536 | 0.3536 |
| 35 | 13.263 | 0.3366 |
| 40 | 14.898 | 0.3171 |
| 45 | 16.429 | 0.2949 |
| 50 | 17.843 | 0.2701 |
| 55 | 19.126 | 0.2428 |
| 60 | 20.267 | 0.2132 |
| 65 | 21.255 | 0.1814 |
| 70 | 22.078 | 0.1476 |
| 75 | 22.728 | 0.1122 |
| 80 | 23.198 | 0.0756 |
| 85 | 23.483 | 0.0380 |
| 90 | 23.578 | 0.0000 |
| 95 | 23.483 | −0.0380 |
| 100 | 23.198 | −0.0756 |
| 105 | 22.728 | −0.1122 |
| 110 | 22.076 | −0.1476 |
| 115 | 21.255 | −0.1814 |
| 120 | 20.267 | −0.2132 |
| 125 | 19.126 | −0.2428 |
| 130 | 17.843 | −0.2701 |
| 135 | 16.429 | −0.2949 |
| 140 | 14.898 | −0.3171 |
| 145 | 13.263 | −0.3366 |
| 150 | 11.536 | −0.3536 |
| 155 | 9.732 | −0.3678 |
| 160 | 7.863 | −0.3794 |
| 165 | 5.942 | −0.3885 |
| 170 | 3.982 | −0.3949 |
| 175 | 1.997 | −0.3987 |
| 180 | 0.000 | −0.4000 |

TABLE V
$R/L=0.5$

| $\theta$, deg | $\phi$, deg | $\phi'/\omega$ |
|---|---|---|
| 0 | 0.000 | 0.5000 |
| 5 | 2.497 | 0.4986 |
| 10 | 4.980 | 0.4948 |
| 15 | 7.435 | 0.4871 |
| 20 | 9.846 | 0.4769 |
| 25 | 12.199 | 0.4636 |
| 30 | 14.477 | 0.4472 |
| 35 | 16.665 | 0.4275 |
| 40 | 18.747 | 0.4045 |
| 45 | 20.704 | 0.3780 |
| 50 | 22.521 | 0.3479 |
| 55 | 24.178 | 0.3144 |
| 60 | 25.658 | 0.2773 |
| 65 | 26.946 | 0.2370 |
| 70 | 28.024 | 0.1937 |
| 75 | 28.879 | 0.1478 |
| 80 | 29.498 | 0.0998 |
| 85 | 29.874 | 0.0503 |
| 90 | 30.000 | 0.0000 |
| 95 | 29.874 | −0.0503 |
| 100 | 29.498 | −0.0998 |
| 105 | 28.879 | −0.1478 |
| 110 | 28.024 | −0.1937 |
| 115 | 26.946 | −0.2370 |
| 120 | 25.658 | −0.2773 |
| 125 | 24.178 | −0.3144 |
| 130 | 22.521 | −0.3479 |
| 135 | 20.704 | −0.3780 |
| 140 | 18.747 | −0.4045 |
| 145 | 16.665 | −0.4275 |
| 150 | 14.477 | −0.4472 |
| 155 | 12.199 | −0.4636 |
| 160 | 9.846 | −0.4769 |
| 165 | 7.435 | −0.4871 |
| 170 | 4.980 | −0.4943 |
| 175 | 2.497 | −0.4986 |
| 180 | 0.000 | −0.5000 |

TABLE VI

$R/L=0.6$

| $\theta$, deg | $\phi$, deg | $\phi'/\omega$ | $\chi'/\omega L$ | $\phi''/\omega^2$ | $\chi''/\omega^2 L$ |
|---|---|---|---|---|---|
| 0 | 0.000 | 0.6000 | −0.0000 | 0.0000 | 0.9600 |
| 5 | 2.997 | 0.5985 | −0.0836 | −0.0336 | 0.9537 |
| 10 | 5.980 | 0.5941 | −0.1661 | −0.0678 | 0.9349 |
| 15 | 8.933 | 0.5867 | −0.2464 | −0.1031 | 0.9036 |
| 20 | 11.841 | 0.5761 | −0.3234 | −0.1401 | 0.8599 |
| 25 | 14.683 | 0.5622 | −0.3961 | −0.1793 | 0.8040 |
| 30 | 17.457 | 0.5447 | −0.4634 | −0.2212 | 0.7363 |
| 35 | 20.129 | 0.5235 | −0.5243 | −0.2661 | 0.6572 |
| 40 | 22.685 | 0.4982 | −0.5778 | −0.3143 | 0.5674 |
| 45 | 25.104 | 0.4685 | −0.6230 | −0.3657 | 0.4679 |
| 50 | 27.363 | 0.4343 | −0.6592 | −0.4199 | 0.3601 |
| 55 | 29.438 | 0.3952 | −0.6857 | −0.4762 | 0.2461 |
| 60 | 31.306 | 0.3511 | −0.7021 | −0.5332 | 0.1283 |
| 65 | 32.941 | 0.3021 | −0.7081 | −0.5888 | 0.0100 |
| 70 | 34.320 | 0.2485 | −0.7039 | −0.6405 | −0.1049 |
| 75 | 35.419 | 0.1906 | −0.6900 | −0.6853 | −0.2123 |
| 80 | 36.219 | 0.1291 | −0.6672 | −0.7202 | −0.3079 |
| 85 | 36.706 | 0.0652 | −0.6367 | −0.7424 | −0.3880 |
| 90 | 36.869 | 0.0000 | −0.6000 | −0.7500 | −0.4500 |
| 95 | 36.706 | −0.0652 | −0.5587 | −0.7423 | −0.4926 |
| 100 | 36.219 | −0.1291 | −0.5146 | −0.7202 | −0.5163 |
| 105 | 35.419 | −0.1906 | −0.4691 | −0.6853 | −0.5229 |
| 110 | 34.320 | −0.2485 | −0.4237 | −0.6405 | −0.5154 |
| 115 | 32.941 | −0.3021 | −0.3795 | −0.5888 | −0.4971 |
| 120 | 31.306 | −0.3511 | −0.3372 | −0.5332 | −0.4717 |
| 125 | 29.438 | −0.3952 | −0.2973 | −0.4762 | −0.4422 |
| 130 | 27.363 | −0.4343 | −0.2600 | −0.4199 | −0.4112 |
| 135 | 25.104 | −0.4685 | −0.2255 | −0.3657 | −0.3806 |
| 140 | 22.685 | −0.4982 | −0.1935 | −0.3142 | −0.3519 |
| 145 | 20.129 | −0.5235 | −0.1640 | −0.2661 | −0.3258 |
| 150 | 17.457 | −0.5447 | −0.1366 | −0.2212 | −0.3029 |
| 155 | 14.688 | −0.5622 | −0.1110 | −0.1793 | −0.2836 |
| 160 | 11.841 | −0.5761 | −0.0870 | −0.1401 | −0.2678 |
| 165 | 8.933 | −0.5867 | −0.0642 | −0.1031 | −0.2556 |
| 170 | 5.980 | −0.5941 | −0.0423 | −0.0678 | −0.2469 |
| 175 | 2.997 | −0.5985 | −0.0210 | −0.0336 | −0.2417 |
| 180 | 0.000 | −0.6000 | −0.0000 | −0.0000 | −0.2400 |

TABLE VII

$R/L=0.7$

| $\theta$, deg | $\phi$, deg | $\phi'/\omega$ | $\chi'/\omega L$ | $\phi''/\omega^2$ | $\chi''/\omega^2 L$ |
|---|---|---|---|---|---|
| 0 | 0.000 | 0.7000 | −0.0000 | 0.0000 | 1.1900 |
| 5 | 3.497 | 0.6986 | −0.1036 | −0.0313 | 1.1826 |
| 10 | 6.981 | 0.6945 | −0.2060 | −0.0634 | 1.1604 |
| 15 | 10.438 | 0.6875 | −0.3057 | −0.0971 | 1.1234 |
| 20 | 13.851 | 0.6775 | −0.4016 | −0.1334 | 1.0719 |
| 25 | 17.207 | 0.6641 | −0.4923 | −0.1731 | 1.0045 |
| 30 | 20.487 | 0.6471 | −0.5765 | −0.2171 | 0.9225 |
| 35 | 23.672 | 0.6261 | −0.6529 | −0.2665 | 0.8254 |
| 40 | 26.740 | 0.6004 | −0.7201 | −0.3222 | 0.7132 |
| 45 | 29.668 | 0.5696 | −0.7769 | −0.3848 | 0.5865 |
| 50 | 32.427 | 0.5331 | −0.8221 | −0.4548 | 0.4459 |
| 55 | 34.988 | 0.4901 | −0.8544 | −0.5318 | 0.2933 |
| 60 | 37.316 | 0.4401 | −0.8730 | −0.6146 | 0.1314 |
| 65 | 39.376 | 0.3827 | −0.8772 | −0.7005 | −0.0354 |
| 70 | 41.131 | 0.3179 | −0.8669 | −0.7851 | −0.2009 |
| 75 | 42.543 | 0.2459 | −0.8424 | −0.8622 | −0.3573 |
| 80 | 43.579 | 0.1678 | −0.8050 | −0.9248 | −0.4956 |
| 85 | 44.213 | 0.0851 | −0.7567 | −0.9659 | −0.6073 |
| 90 | 44.427 | 0.0000 | −0.7000 | −0.9802 | −0.6861 |
| 95 | 44.213 | −0.0851 | −0.6380 | −0.9659 | −0.7293 |
| 100 | 43.579 | −0.1678 | −0.5737 | −0.9248 | −0.7387 |
| 105 | 42.543 | −0.2459 | −0.5099 | −0.8622 | −0.7196 |
| 110 | 41.131 | −0.3179 | −0.4487 | −0.7851 | −0.6797 |
| 115 | 39.376 | −0.3827 | −0.3916 | −0.7005 | −0.6270 |
| 120 | 37.316 | −0.4401 | −0.3394 | −0.6146 | −0.5686 |
| 125 | 34.988 | −0.4901 | −0.2924 | −0.5318 | −0.5098 |
| 130 | 32.427 | −0.5331 | −0.2504 | −0.4548 | −0.4539 |
| 135 | 29.668 | −0.5696 | −0.2130 | −0.3848 | −0.4035 |
| 140 | 26.740 | −0.6004 | −0.1798 | −0.3221 | −0.3592 |
| 145 | 23.672 | −0.6261 | −0.1501 | −0.2665 | −0.3214 |
| 150 | 20.487 | −0.6471 | −0.1235 | −0.2171 | −0.2899 |
| 155 | 17.207 | −0.6641 | −0.0994 | −0.1731 | −0.2643 |
| 160 | 13.851 | −0.6775 | −0.0772 | −0.1334 | −0.2441 |
| 165 | 10.438 | −0.6875 | −0.0566 | −0.0971 | −0.2289 |
| 170 | 6.981 | −0.6945 | −0.0371 | −0.0634 | −0.2183 |
| 175 | 3.497 | −0.6986 | −0.0184 | −0.0313 | −0.2120 |
| 180 | 0.000 | −0.7000 | −0.0000 | −0.0000 | −0.2100 |

| $\chi'/\omega L$ | $\phi''/\omega^2$ | $\chi''/\omega^2 L$ |
|---|---|---|
| −0.0000 | 0.0000 | 0.5600 |
| −0.0488 | −0.0293 | 0.5563 |
| −0.0969 | −0.0588 | 0.5454 |
| −0.1437 | −0.0884 | 0.5273 |
| −0.1887 | −0.1182 | 0.5023 |
| −0.2312 | −0.1483 | 0.4708 |
| −0.2707 | −0.1786 | 0.4332 |
| −0.3067 | −0.2090 | 0.3900 |
| −0.3386 | −0.2393 | 0.3420 |
| −0.3662 | −0.2692 | 0.2901 |
| −0.3892 | −0.2984 | 0.2351 |
| −0.4072 | −0.3264 | 0.1782 |
| −0.4203 | −0.3525 | 0.1205 |
| −0.4283 | −0.3762 | 0.0633 |
| −0.4314 | −0.3968 | 0.0079 |
| −0.4297 | −0.4136 | −0.0447 |
| −0.4237 | −0.4261 | −0.0931 |
| −0.4136 | −0.4338 | −0.1367 |
| −0.4000 | −0.4364 | −0.1746 |
| −0.3833 | −0.4338 | −0.2064 |
| −0.3642 | −0.4261 | −0.2321 |
| −0.3430 | −0.4136 | −0.2517 |
| −0.3204 | −0.3968 | −0.2658 |
| −0.2968 | −0.3762 | −0.2748 |
| −0.2726 | −0.3525 | −0.2795 |
| −0.2481 | −0.3264 | −0.2806 |
| −0.2237 | −0.2984 | −0.2791 |
| −0.1994 | −0.2692 | −0.2756 |
| −0.1756 | −0.2393 | −0.2708 |
| −0.1522 | −0.2090 | −0.2653 |
| −0.1293 | −0.1786 | −0.2597 |
| −0.1069 | −0.1483 | −0.2543 |
| −0.0849 | −0.1182 | −0.2494 |
| −0.0633 | −0.0884 | −0.2454 |
| −0.0420 | −0.0588 | −0.2425 |
| −0.0210 | −0.0293 | −0.2406 |
| −0.0000 | −0.0000 | −0.2400 |

| $\chi'/\omega L$ | $\phi''/\omega^2$ | $\chi''/\omega^2 L$ |
|---|---|---|
| −0.0000 | 0.0000 | 0.7500 |
| −0.0653 | −0.0328 | 0.7450 |
| −0.1297 | −0.0658 | 0.7301 |
| −0.1924 | −0.0995 | 0.7053 |
| −0.2526 | −0.1341 | 0.6710 |
| −0.3093 | −0.1697 | 0.6274 |
| −0.3618 | −0.2066 | 0.5750 |
| −0.4094 | −0.2446 | 0.5145 |
| −0.4514 | −0.2839 | 0.4467 |
| −0.4872 | −0.3240 | 0.3726 |
| −0.5163 | −0.3644 | 0.2936 |
| −0.5383 | −0.4046 | 0.2112 |
| −0.5531 | −0.4434 | 0.1273 |
| −0.5606 | −0.4798 | 0.0440 |
| −0.5609 | −0.5123 | −0.0366 |
| −0.5543 | −0.5395 | −0.1120 |
| −0.5415 | −0.5601 | −0.1803 |
| −0.5231 | −0.5730 | −0.2397 |
| −0.5000 | −0.5773 | −0.2887 |
| −0.4731 | −0.5730 | −0.3268 |
| −0.4433 | −0.5601 | −0.3540 |
| −0.4116 | −0.5395 | −0.3709 |
| −0.3788 | −0.5123 | −0.3786 |
| −0.3457 | −0.4798 | −0.3786 |
| −0.3129 | −0.4434 | −0.3727 |
| −0.2808 | −0.4046 | −0.3623 |
| −0.2497 | −0.3644 | −0.3492 |
| −0.2199 | −0.3240 | −0.3343 |
| −0.1914 | −0.2839 | −0.3193 |
| −0.1642 | −0.2446 | −0.3046 |
| −0.1382 | −0.2066 | −0.2910 |
| −0.1133 | −0.1697 | −0.2789 |
| −0.0895 | −0.1341 | −0.2687 |
| −0.0664 | −0.0995 | −0.2606 |
| −0.0439 | −0.0659 | −0.2547 |
| −0.0219 | −0.0328 | −0.2512 |
| −0.0000 | −0.0000 | −0.2500 |

$$\phi' = (251.33)(0.2495) = 62.68 \text{ rad/sec}$$
$$\phi'' = (251.33)^2(0.1638) = 10{,}347 \text{ rad/sec}^2$$
$$x' = (12)(251.33)(0.2150) = 648.4 \text{ in./sec}$$
$$x'' = (12)(63{,}167)(0.2789) = 211{,}380 \text{ in./sec}^2$$

$$Mr\phi'' = \left(\frac{8.106}{386}\right)(6.4)(10{,}347) = 1391 \text{ lb}$$

$$\sin\phi = 0.1719; \quad \cos\phi = 0.9851$$

$$Mx'' \sin\phi = \left(\frac{8.106}{386}\right)(211{,}380)(0.1719) = 763.2 \text{ lb}$$

Substituting these values into Eq 13 gives

$$F_R = 1391 + 763 = 2154 \text{ lb}$$

Substituting into Eq 16

$$d = \frac{(1.109)(10{,}347)+(763.2)(6.4)}{2154} = 7.59 \text{ in.}$$

$$Mr(\phi')^2 = (0.021)(6.4)(62.68)^2 = 528.1 \text{ lb}$$
$$Mx'' \cos\phi = (4439)(0.9851) = 4373 \text{ lb}$$

Substituting into Eq 17

$$F_L = 528 + 4373 = 4901 \text{ lb}$$

**Part B:** Find the bending and tensile stresses caused by the inertia forces.

The connecting rod, Fig 4, is assumed to be a trapezoidal steel bar of rectangular cross section of constant thickness, $t$, with a width of 3 in. at the crank pin and a width of 2 in. at the wrist pin end. Also, density of steel $\rho = 0.28$ lb/in.$^3$

Let $A$ = constant = $\rho t/g$ = $(0.28)(0.965)/386$ = 0.0007 lb-sec$^2$/in.$^3$. Also from Fig 4

$$h = 2 + y/12$$

Then
$$I_C = A\int_0^{12} hy^2\,dy$$
$$= 0.0007\left[\frac{2y^3}{3} + \frac{y^4}{48}\right]_0^{12}$$
$$I_C = 1.109 \text{ in.-lb-sec}^2$$

and
$$K^2 = \frac{1.109}{0.021} = 52.8 \text{ in.}^2$$

Referring to the load diagram in Fig 4, the connecting rod can be considered as a simply supported beam loaded with a distributed varying load $W$. The load $W$ at any point $y$ is

$$W = A\int h\,(y\phi'' + x''\sin\phi)\,dy$$
$$A\int\left(2y\phi'' + \frac{y^2}{12}\phi'' + 2x''\sin\phi + \frac{y}{12}x''\sin\phi\right)dy$$

The reaction $R_C$ is

$$R_C = F_R(L - d)/L$$

The bending moment at any point $z$ is

$$M_b = -zR_C + A\int_0^z\left[2y\phi'' + \frac{y^2}{12}\phi'' + 2x''\sin\phi + \frac{y}{12}x''\sin\phi\right](z - y)\,dy$$

Perform the indicated integration:

$$M_b = -zR_C + A\left[\frac{z^3}{3}\phi'' + \frac{z^4}{144}\phi'' + z^2x''\sin\phi + \frac{z^3}{72}x''\sin\phi\right]$$

The maximum bending moment occurs where $dM_b/dz = 0$. Hence, differentiating the above equation gives

$$\frac{dM_b}{dz} = 0 = -R_C + A\left[z^2\phi'' + \frac{z^3}{36}\phi'' + 2zx''\sin\phi + \frac{z^2}{24}x''\sin\phi\right]$$

Factor $z$ is the distance from $C$ to the point of maximum bending moment. To determine the crank angle at which point the maximum bending moment occurs, values for the resulting force, $F_R$, and the moment of the resulting force, $F_Rd$, about the wrist pin are tabulated in Table VIII for 30-deg increments of $\theta$. (This requires repeating the calculations in Part A for every 30 deg, hence the value of the design tables.) The maximum moment occurs at $\theta = 240$ deg.

Using the appropriate values for this angle gives

$$\frac{R_C}{M\omega^2} = \frac{F_R}{M\omega^2}\left[\frac{12 - d}{12}\right]$$
$$= (2.288)\left[\frac{12 - 7.76}{12}\right] = 0.8084$$

Therefore

$$R_C/\omega^2 = \left(\frac{8.106}{386}\right)(0.8084) = 0.0169$$

From which

$$z^3 + 39.47z^2 + 166.77z - 3326.1 = 0$$

This cubic equation has three real roots, only one of which is positive. The two negative roots have no physical meaning in this case. Solving the cubic equation results in

$$z = 6.862$$

This value of the root is substituted into the bending moment equation to find the maximum bending moment in the connecting rod when the angle $\theta = 240$ deg from the top dead center:

$$M_b = 4540 \text{ in.-lb.}$$

At the section where the bending moment is maximum

$$h = 2 + 0.569 = 2.569 \text{ in.}$$
$$I = 0.965\,(2.569)^3/12 = 1.363 \text{ in.}^4$$

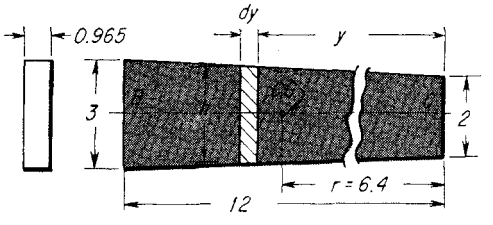
Bending stress equals

$$S_b = \frac{M_bh}{2I} = \frac{(4540)(1.284)}{1.364}$$
$$S_b = 4270 \text{ psi}$$

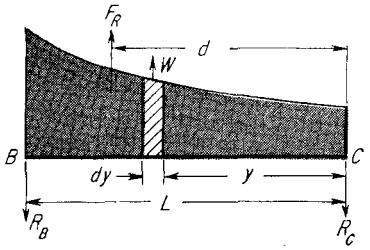
**Example II—mechanical press**

Stroke = 16 in.; connecting rod = 26.67 in. When crank is 25 deg from the bottom dead center, Fig 5, the torque exerted by flywheel on crankshaft is 7500 ft-lb. Thus $R/L = 0.3$, and $F_t = 7500\,(12)/8 = 11{,}250$.

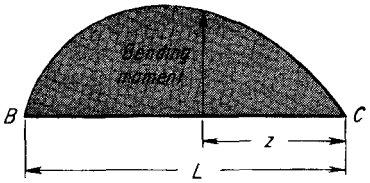
## 4.. Load and bending-moment analysis




(A) Simplified section of connecting rod




(B) Load diagram

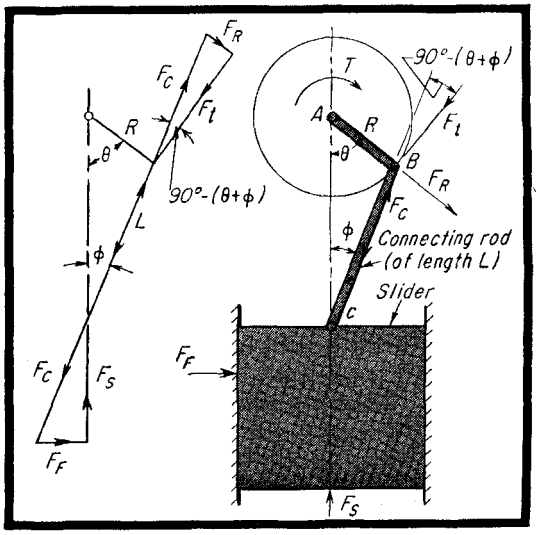



(C) Bending moment diagram

TABLE **VIII.. Dynamic forces for $R/L = 0.3$**

| $\theta$ | $\phi$ | $\phi''/\omega^2$ | $\chi''/L\omega^2$ | $F_R/M\omega^2$ | d | $F_R d/M\omega^2$ |
|---|---|---|---|---|---|---|
| 30 | 8.626 | 0.1412 | +0.3069 | 1.4567 | 7.53 | 10.97 |
| 60 | 15.058 | 0.2626 | +0.1050 | 2.0081 | 7.95 | 15.96 |
| 75 | 16.844 | 0.3007 | −0.0032 | 1.9137 | 8.26 | 15.81 |
| 90 | 17.457 | 0.3145 | −0.0943 | 1.6730 | 8.60 | 14.40 |
| 120 | 15.058 | 0.2626 | −0.1949 | 1.0723 | 9.30 | 9.97 |
| 150 | 8.626 | 0.1412 | −0.2127 | 0.5208 | 9.61 | 5.00 |
| 180 | 0.000 | 0.0000 | −0.2100 | 0.0000 | .. | 0.00 |
| 210 | 8.626 | 0.1412 | +0.2127 | 1.2871 | 7.70 | 9.91 |
| 240 | 15.058 | 0.2626 | +0.1949 | 2.2883 | 7.76 | 17.75 |
| 270 | 17.457 | 0.3145 | +0.0943 | 2.3524 | 7.98 | 18.78 |
| 285 | 16.844 | 0.3008 | +0.0032 | 2.0361 | 8.24 | 15.95 |
| 300 | 15.058 | 0.2626 | −0.1050 | 1.3525 | 8.72 | 11.76 |
| 330 | 8.626 | 0.1412 | −0.3068 | 0.3511 | 11.16 | 3.92 |

## 5.. Forces on punch press




From Table III $\phi = 7.283$ deg; hence $\phi + \theta = 32.283$ deg. The forces exerted by the slide and connection are

$$F_s = \frac{F_t \cos \phi}{\sin (\theta + \phi)} \qquad F_C = \frac{F_t}{\sin (\theta + \phi)}$$

Hence

$$F_s = \frac{(11{,}250)(0.9919)}{0.5341} = 20{,}900 \text{ lb.}$$

$$F_C = \frac{11{,}250}{0.5341} = 21{,}100 \text{ lb.}$$

The radial force on the crank pin is

$$F_R = \frac{F_t \cos (\theta + \phi)}{\sin (\theta + \phi)}$$

$$F_R = \frac{(11{,}250)(0.8454)}{0.5341} = 17{,}800 \text{ lb.}$$

The force of the slide against the frame is

$$F_F = \frac{F_t \sin \phi}{\sin (\theta + \phi)}$$

$$F_F = \frac{(11{,}250)(0.1268)}{0.5341} = 2675 \text{ lb.}$$

The value for the side thrust is useful in estimating the wear rate of the guides.

Now, assuming the press makes three working strokes per minute, its angular speed is

$$\omega = 3(2\pi)/60 = 0.3142 \text{ rad/sec}$$

From Table III, the relative angular velocity, $\phi'$, of the connecting-rod knuckle to its bearing is

$$\phi' = (0.3142)(0.2741) = 0.0861 \text{ rad/sec}$$

The relative angular velocity of the crank pin to its bearing (at point $B$) is

$$\theta' + \phi' = 0.3142 + 0.0861 = 0.4003 \text{ rad/sec}$$

These velocities are then used to estimate the wear life of the knuckle and crank-pin bearings.

# Torque Factors for Slide Crank Mechanisms

Alexander Hammer



THE ACCOMPANYING TABLE gives values of a factor $f_t$ for determining torque imposed on a crankshaft by a force acting on a crosshead in the direction of stroke.

- $P$ = force on piston or crosshead, lb.
- $\theta$ = crank position, from head end dead center, deg
- $\beta$ = connecting rod angle corresponding to $\theta$, deg
- $P_l$ = force on connecting rod, lb
- $P_t = P/\cos\beta$, lb
- $l$ = connecting rod length, in.
- $r$ = crank radius, in.
- $k = l/r$ = ratio of length of connecting rod to radius of crank
- $a$ = perpendicular distance from centerline of connecting rod to center of crank shaft, in.
- $a = r \sin(\theta + \beta)$, in.
- $T$ = torque on crankshaft, lb in.
- $f_t$ = factor from accompanying table for specific $k$ and crank angle $\theta$
- $T = P_t a = (P/\cos\beta)\, r \sin(\theta + \beta)$

$$= Pr\left[\frac{\sin\theta\cos\beta + \cos\theta\sin\beta}{\cos\beta}\right] \qquad (1)$$

since $\frac{r}{kr} = \frac{\sin\beta}{\sin\theta}$, then $\sin\beta = \frac{\sin\theta}{k}$

and $\sqrt{1 - \cos^2\beta} = \sin\theta/k$

or $\cos\beta = \sqrt{k^2 - \sin^2\theta}/k$

substituting these values of $\sin\beta$ and $\cos\beta$ in Eq (1), there results

$$T = Pr\left[\sin\theta\left(1 + \frac{\cos\theta}{\sqrt{k^2 - \sin^2\theta}}\right)\right] \qquad (2)$$

substituting $f_t$ for the expression contained in the brackets of Eq (2), there results

$$T = Prf_t \qquad (3)$$

Values of $f_t$ are given in the accompanying table for $k$ ratios ranging from 3.0 to 5.0.

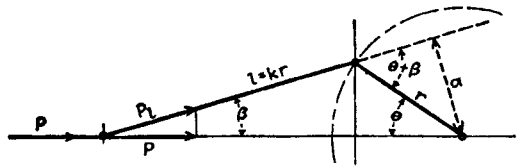
PROBLEM: Find torque on crankshaft resulting from a force of 150 lb on crosshead, when crank radius is 3 in., connecting rod length is 12 6 in., and crank angle is 40 deg from head end dead center?

SOLUTION:

$l = 12.6$, $r = 3$, $k = l/r = 12.6/3 = 4.2$

From Table, for $k$ equal to 4.2 and $\theta$ equal to 40 deg, $f_t$ is 0.7614, then

$T = Prf_t = 150 \times 3 \times 0.7614 = 342.63$ lb in.




## Tangential Factors $f_t$

| Crank angle deg. | Ratio of Length of Rod to Length of Crank $l/r = k$ | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3.0 | 3.1 | 3.2 | 3.3 | 3.4 | 3.5 | 3.6 | 3.7 | 3.8 | 3.9 | 4.0 | 4.1 | 4.2 | 4.3 | 4.4 | 4.5 | 4.6 | 4.7 | 4.8 | 4.9 | 5.0 |
| 0 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 |
| 5 | 0.1161 | 0.1152 | 0.1143 | 0.1135 | 0.1127 | 0.1120 | 0.1113 | 0.1106 | 0.1100 | 0.1094 | 0.1089 | 0.1083 | 0.1078 | 0.1073 | 0.1069 | 0.1064 | 0.1060 | 0.1056 | 0.1052 | 0.1049 | 0.1045 |
| 10 | 0.2308 | 0.2289 | 0.2272 | 0.2255 | 0.2240 | 0.2226 | 0.2212 | 0.2199 | 0.2187 | 0.2175 | 0.2164 | 0.2154 | 0.2144 | 0.2135 | 0.2125 | 0.2117 | 0.2109 | 0.2101 | 0.2093 | 0.2086 | 0.2079 |
| 15 | 0.3425 | 0.3397 | 0.3372 | 0.3348 | 0.3326 | 0.3304 | 0.3284 | 0.3266 | 0.3248 | 0.3231 | 0.3215 | 0.3199 | 0.3185 | 0.3171 | 0.3157 | 0.3145 | 0.3133 | 0.3121 | 0.3110 | 0.3099 | 0.3089 |
| 20 | 0.4499 | 0.4463 | 0.4430 | 0.4399 | 0.4370 | 0.4343 | 0.4317 | 0.4293 | 0.4269 | 0.4247 | 0.4227 | 0.4207 | 0.4188 | 0.4170 | 0.4153 | 0.4136 | 0.4121 | 0.4106 | 0.4091 | 0.4078 | 0.4064 |
| 25 | 0.5516 | 0.5473 | 0.5434 | 0.5397 | 0.5362 | 0.5329 | 0.5298 | 0.5268 | 0.5240 | 0.5214 | 0.5189 | 0.5165 | 0.5143 | 0.5121 | 0.5101 | 0.5081 | 0.5062 | 0.5044 | 0.5027 | 0 5011 | 0 4995 |
| 30 | 0.6464 | 0.6415 | 0.6370 | 0.6328 | 0.6288 | 0.6250 | 0.6215 | 0.6181 | 0.6150 | 0.6120 | 0 6091 | 0.6064 | 0.6038 | 0.6014 | 0.5991 | 0.5968 | 0.5947 | 0.5927 | 0.5907 | 0.5888 | 0 5870 |
| 35 | 0.7331 | 0.7278 | 0.7228 | 0.7182 | 0.7138 | 0.7097 | 0.7058 | 0.7021 | 0.6987 | 0.6954 | 0 6923 | 0.6893 | 0.6865 | 0.6838 | 0.6813 | 0.6788 | 0 6765 | 0.6743 | 0.6722 | 0.6701 | 0.6682 |
| 40 | 0.8108 | 0.8052 | 0.7999 | 0.7949 | 0.7903 | 0.7859 | 0.7818 | 0.7779 | 0.7743 | 0.7708 | 0.7675 | 0.7644 | 0.7614 | 0.7586 | 0.7559 | 0.7533 | 0.7509 | 0.7486 | 0.7463 | 0 7442 | 0 7421 |
| 45 | 0.8786 | 0.8728 | 0.8673 | 0.8622 | 0.8575 | 0.8530 | 0.8488 | 0.8448 | 0.8410 | 0.8375 | 0.8341 | 0.8309 | 0.8279 | 0.8250 | 0.8222 | 0.8196 | 0.8171 | 0.8147 | 0.8124 | 0.8102 | 0.8081 |
| 50 | 0.9358 | 0.9300 | 0 9245 | 0.9194 | 0.9147 | 0.9102 | 0.9060 | 0.9021 | 0.8983 | 0.8948 | 0.8915 | 0.8883 | 0.8853 | 0.8824 | 0.8797 | 0.8771 | 0.8746 | 0.8722 | 0.8700 | 0.8678 | 0.8657 |
| 55 | 0.9820 | 0.9763 | 0.9710 | 0.9661 | 0.9615 | 0 9572 | 0.9532 | 0 9494 | 0.9458 | 0.9424 | 0.9392 | 0.9361 | 0.9332 | 0.9305 | 0.9278 | 0.9253 | 0.9230 | 0.9207 | 0.9185 | 0.9164 | 0.9144 |
| 60 | 1.0168 | 1.0115 | 1.0066 | 1.0020 | 0 9977 | 0 9937 | 0.9900 | 0.9864 | 0.9831 | 0.9799 | 0.9769 | 0.9741 | 0.9714 | 0.9688 | 0.9664 | 0.9641 | 0.9619 | 0.9598 | 0.9577 | 0.9558 | 0.9540 |
| 65 | 1.0402 | 1.0355 | 1.0311 | 1.0270 | 1.0232 | 1.0196 | 1.0162 | 1.0131 | 1.0101 | 1.0073 | 1.0046 | 1.0021 | 0.9997 | 0.9974 | 0.9953 | 0.9932 | 0.9912 | 0.9894 | 0.9876 | 0 9859 | 0.9842 |
| 70 | 1.0525 | 1.0485 | 1.0448 | 1.0413 | 1.0380 | 1.0350 | 1.0322 | 1.0295 | 1.0270 | 1.0246 | 1.0224 | 1.0202 | 1.0182 | 1.0163 | 1.0145 | 1.0127 | 1.0111 | 1.0095 | 1.0080 | 1.0065 | 1.0051 |
| 75 | 1.0539 | 1.0508 | 1.0479 | 1.0451 | 1.0426 | 1.0402 | 1.0380 | 1.0359 | 1.0339 | 1.0321 | 1.0303 | 1.0287 | 1 0271 | 1.0256 | 1.0242 | 1.0228 | 1.0215 | 1.0203 | 1.0191 | 1.0180 | 1 0169 |
| 80 | 1.0452 | 1.0430 | 1.0410 | 1.0391 | 1.0374 | 1.0357 | 1.0342 | 1.0328 | 1.0314 | 1.0301 | 1.0289 | 1.0278 | 1.0267 | 1.0257 | 1.0247 | 1.0238 | 1.0229 | 1.0220 | 1.0212 | 1.0204 | 1.0197 |
| 85 | 1.0269 | 1.0258 | 1.0247 | 1.0238 | 1.0229 | 1.0221 | 1.0213 | 1.0206 | 1.0199 | 1.0192 | 1.0186 | 1.0180 | 1.0175 | 1.0169 | 1.0164 | 1.0160 | 1.0155 | 1 0151 | 1.0147 | 1.0143 | 1.0139 |
| 90 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1 0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1 0000 | 1.0000 | 1.0000 | 1 0000 |
| 95 | 0.9655 | 0.9666 | 0.9676 | 0.9686 | 0.9695 | 0.9703 | 0.9711 | 0.9718 | 0.9725 | 0.9732 | 0.9738 | 0.9744 | 0.9749 | 0.9754 | 0.9759 | 0.9764 | 0.9769 | 0.9773 | 0.9777 | 0.9781 | 0.9785 |
| 100 | 0.9245 | 0.9266 | 0.9286 | 0.9305 | 0.9323 | 0.9339 | 0.9354 | 0.9369 | 0.9382 | 0.9395 | 0.9407 | 0.9418 | 0.9429 | 0.9440 | 0.9449 | 0.9459 | 0.9468 | 0.9476 | 0.9484 | 0.9492 | 0 9499 |
| 105 | 0.8779 | 0.8810 | 0.8840 | 0.8867 | 0.8892 | 0 8916 | 0.8938 | 0.8959 | 0 8979 | 0.8998 | 0 9015 | 0 9032 | 0.9048 | 0.9063 | 0.9077 | 0.9090 | 0.9103 | 0.9116 | 0.9127 | 0.9139 | 0.9150 |
| 110 | 0.8269 | 0.8309 | 0.8346 | 0.8381 | 0.8413 | 0.8444 | 0.8472 | 0.8499 | 0 8524 | 0.8548 | 0 8570 | 0.8592 | 0.8612 | 0 8631 | 0 8649 | 0.8667 | 0.8683 | 0.8699 | 0.8714 | 0.8729 | 0.8742 |
| 115 | 0.7724 | 0.7771 | 0.7815 | 0.7856 | 0 7894 | 0.7930 | 0 7964 | 0 7995 | 0.8025 | 0 8053 | 0 8080 | 0.8105 | 0.8129 | 0.8152 | 0 8174 | 0.8194 | 0.8214 | 0.8233 | 0.8251 | 0.8268 | 0.8284 |
| 120 | 0.7153 | 0.7206 | 0.7255 | 0.7300 | 0.7343 | 0.7383 | 0.7421 | 0.7457 | 0.7490 | 0.7522 | 0.7551 | 0.7580 | 0.7607 | 0.7632 | 0.7657 | 0.7680 | 0.7702 | 0.7723 | 0.7743 | 0.7762 | 0.7781 |
| 125 | 0.6563 | 0.6620 | 0.6673 | 0.6722 | 0.6768 | 0.6811 | 0.6851 | 0.6889 | 0.6925 | 0.6959 | 0.6991 | 0.7022 | 0.7051 | 0.7078 | 0.7105 | 0.7130 | 0.7154 | 0.7176 | 0.7198 | 0.7219 | 0.7239 |
| 130 | 0.5963 | 0.6021 | 0.6076 | 0.6126 | 0.6174 | 0 6219 | 0 6261 | 0 6300 | 0.6337 | 0.6373 | 0.6406 | 0.6438 | 0.6468 | 0.6497 | 0.6524 | 0.6550 | 0.6575 | 0.6599 | 0.6621 | 0.6643 | 0.6664 |
| 135 | 0.5356 | 0 5415 | 0.5469 | 0.5520 | 0.5568 | 0 5612 | 0.5655 | 0.5694 | 0 5732 | 0.5767 | 0 5801 | 0.5833 | 0.5863 | 0.5892 | 0.5920 | 0.5946 | 0.5971 | 0.5995 | 0.6018 | 0.6040 | 0.6061 |
| 140 | 0.4748 | 0 4804 | 0 4857 | 0.4907 | 0.4953 | 0 4997 | 0.5038 | 0.5077 | 0 5113 | 0.5148 | 0 5181 | 0.5212 | 0.5242 | 0.5270 | 0.5297 | 0.5322 | 0.5347 | 0.5370 | 0 5393 | 0.5414 | 0.5435 |
| 145 | 0.4140 | 0.4194 | 0.4243 | 0.4290 | 0.4334 | 0.4375 | 0.4414 | 0.4450 | 0.4485 | 0.4518 | 0.4549 | 0.4578 | 0.4607 | 0.4633 | 0.4659 | 0.4683 | 0.4706 | 0.4729 | 0.4750 | 0.4770 | 0.4790 |
| 150 | 0.3536 | 0.3585 | 0.3630 | 0 3673 | 0.3712 | 0.3750 | 0.3785 | 0.3819 | 0.3850 | 0.3880 | 0.3909 | 0 3936 | 0.3962 | 0.3986 | 0.4009 | 0.4032 | 0.4053 | 0.4073 | 0.4093 | 0.4112 | 0.4130 |
| 155 | 0.2937 | 0.2979 | 0.3019 | 0.3056 | 0.3091 | 0 3124 | 0.3155 | 0.3184 | 0.3212 | 0 3238 | 0.3263 | 0.3287 | 0.3310 | 0.3331 | 0.3352 | 0.3371 | 0.3390 | 0.3408 | 0.3425 | 0.3442 | 0.3457 |
| 160 | 0.2342 | 0.2377 | 0.2410 | 0.2441 | 0.2470 | 0 2498 | 0.2523 | 0.2548 | 0 2571 | 0.2593 | 0.2614 | 0.2634 | 0.2652 | 0.2670 | 0.2688 | 0.2704 | 0.2720 | 0.2735 | 0.2749 | 0.2763 | 0.2776 |
| 165 | 0.1752 | 0.1779 | 0.1804 | 0.1828 | 0.1851 | 0.1872 | 0.1892 | 0.1911 | 0.1929 | 0.1946 | 0.1962 | 0.1977 | 0.1992 | 0.2006 | 0.2019 | 0.2032 | 0.2044 | 0.2055 | 0.2067 | 0.2077 | 0.2088 |
| 170 | 0.1166 | 0.1184 | 0 1201 | 0.1218 | 0.1233 | 0.1247 | 0.1261 | 0.1274 | 0.1286 | 0.1298 | 0.1309 | 0.1319 | 0.1329 | 0.1338 | 0.1348 | 0.1356 | 0.1364 | 0.1372 | 0.1380 | 0.1387 | 0.1394 |
| 175 | 0.0582 | 0.0591 | 0.0600 | 0.0608 | 0.0616 | 0.0623 | 0.0630 | 0.0637 | 0.0643 | 0.0649 | 0.0654 | 0.0660 | 0.0665 | 0.0670 | 0.0674 | 0.0679 | 0.0683 | 0.0687 | 0.0691 | 0.0694 | 0.0698 |
| 180 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 | 0.0000 |

# Mechanisms for Producing Specific Types of Motions

Sigmund Rappaport

## Straight Line Motion

FIG. 1—No linkages or guides are used in this modified hypocyclic drive which is relatively small in relation to the length of its stroke. The sun gear of pitch diameter $D$ is stationary. The drive shaft, which turns the T-shaped arm, is concentric with this gear. The idler and planet gears, the latter having a pitch diameter of $D/2$, rotate freely on pivots in the arm extensions. Pitch diameter of the idler is of no geometrical significance, although this gear does have an important mechanical function. It reverses the rotation of the planet gear, thus producing true hypocyclic motion with ordinary spur gears only. Such an arrangement occupies only about half as much space as does an equivalent mechanism containing an internal gear. Center distance $R$ is the sum of $D/2$, $D/4$ and an arbitrary distance $d$, determined by a particular application. Points $A$ and $B$ on the driven link, which is fixed to the planet, describe straight-line paths through a stroke of $4R$. All points between $A$ and $B$ trace ellipses, while the line $AB$ envelopes an astroid.

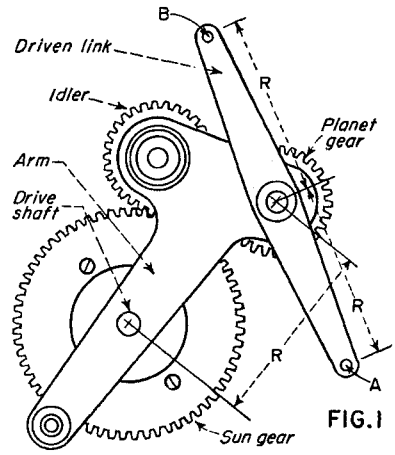



FIG. 1

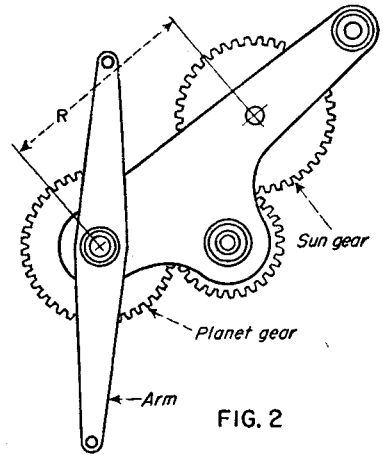
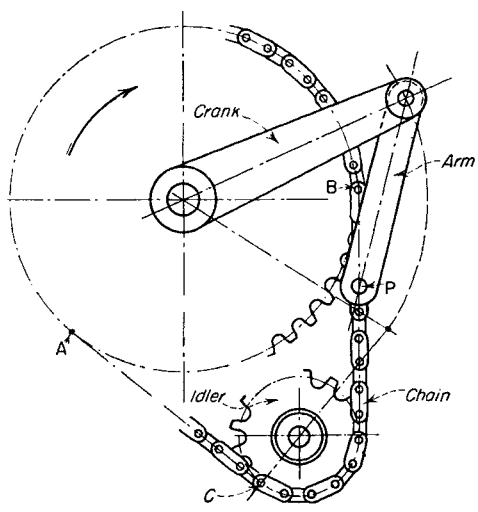



FIG. 2

## Parallel Motion

FIG. 2—A slight modification of the mechanism in Fig. 1 will produce another type of useful motion. If the planet gear has the same diameter as that of the sum gear, the arm will remain parallel to itself throughout the complete cycle. All points on the arm will thereby describe circles of radius $R$. Here again, the position and diameter of the idler gear are of no geometrical importance. This mechanism can be used, for example, to cross-perforate a uniformly moving paper web. The value for $R$ is chosen such that $2\pi R$, or the circumference of the circle described by the needle carrier, equals the desired distance between successive lines of perforations. If the center distance $R$ is made adjustable, the spacing of perforated lines can be varied as desired.



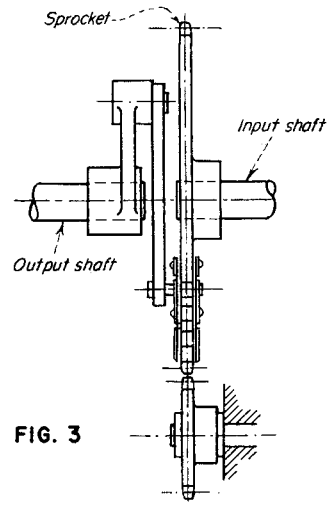




FIG. 3

## Intermittent Motion

FIG. 3—This mechanism, developed by the author and to his knowledge novel, can be adapted to produce a stop, a variable speed without stop or a variable speed with momentary reverse motion. Uniformly rotating input shaft drives the chain around the sprocket and idler, the arm serving as a link between the chain and the end of the output shaft crank. The sprocket drive must be in the ratio $N/n$ with the cycle of the machine, where $n$ is the number of teeth on the sprocket and $N$ the number of links in the chain. When point $P$ travels around the sprocket from point $A$ to position $B$, the crank rotates uniformly. Between $B$ and $C$, $P$ decelerates; between $C$ and $A$ it accelerates;; and at $C$ there is a momentary dwell. By changing the size and position of the idler, or the lengths of the arm and crank, a variety of motions can be obtained. If in the sketch, the length of the crank is shortened, a brief reverse period will occur in the vicinity of $C$; if the crank is lengthened, the output velocity will vary between a maximum and minimum without reaching zero.

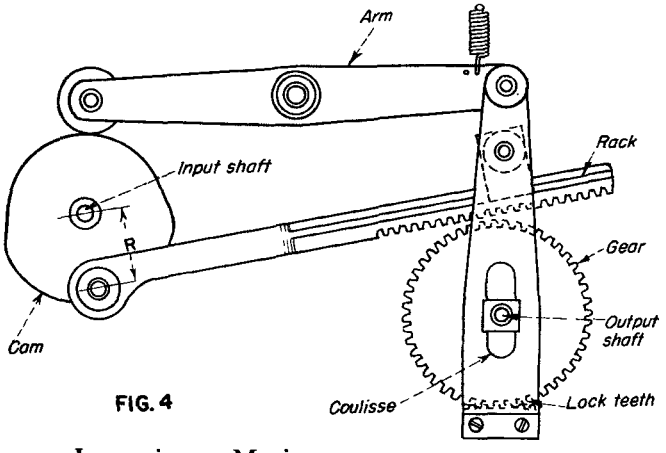



FIG. 4

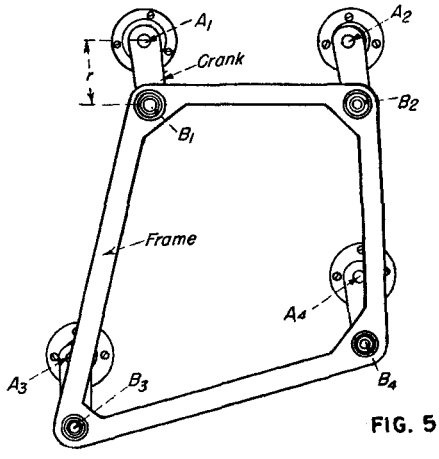



FIG. 5

## Intermittent Motion

FIG. 4—An operating cycle of 180 deg motion and 180 deg dwell is produced by this mechanism. The input shaft drives the rack which is engaged with the output shaft gear during half the cycle. When the rack engages, the lock teeth at the lower end of the coulisse are disengaged and, conversely, when the rack is disengaged, the coulisse teeth are engaged, thereby locking the output shaft positively. The change-over points occur at the dead-center positions so that the motion of the gear is continuously and positively governed. By varying $R$ and the diameter of the gear, the number of revolutions made by the output shaft during the operating half of the cycle can be varied to suit requirements.

## Rotational Motion

FIG. 5—The absence of backlash makes this old but little used mechanism a precision, low-cost replacement for gear or chain drives otherwise used to rotate parallel shafts. Any number of shafts greater than two can be driven from any one of the shafts, provided two conditions are fulfilled: (1) All cranks must have the same length $r$; and (2) the two polygons formed by the shafts $A$ and frame pivot centers $B$ must be identical. The main disadvantage of this mechanism is its dynamic unbalance, which limits the speed of rotation. To lessen the effect of the vibrations produced, the frame should be made as light as is consistent with strength requirements.

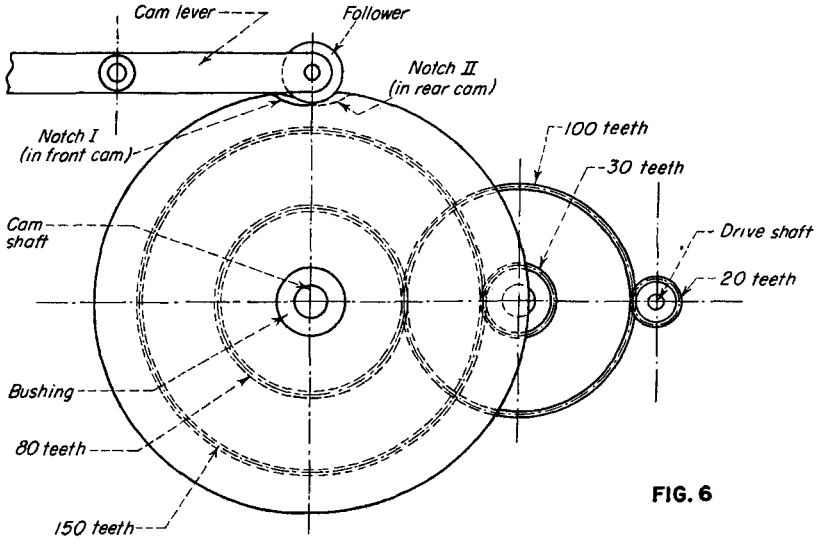



FIG. 6

## Fast Cam-Follower Motion

FIG. 6—Fast cam action every $n$ cycles when $n$ is a relatively large number, can be obtained with this manifold cam and gear mechanism. A single notched cam geared $1/n$ to a shaft turning once a cycle moves relatively slowly under the follower. The double notched-cam arrangement shown is designed to operate the lever once in 100 cycles, imparting to it a rapid movement. One of the two identical cams and the 150-tooth gear are keyed to the bushing which turns freely around the cam shaft. The latter carries the second cam and the 80-tooth gear. The 30- and 100-tooth gears are integral, while the 20-tooth gear is attached to the one-cycle drive shaft. One of the cams turns in the ratio of 20/80 or 1/4; the other in the ratio 20/100 times 30/150 or 1/25. The notches therefore coincide once every 100 cycles ($4 \times 25$). Lever movement is the equivalent of a cam turning in a ratio of 1/4 in relation to the drive shaft. To obtain fast cam action, $n$ must be broken down into prime factors. For example, if 100 were factored into 5 and 20, the notches would coincide after every 20 cycles.

# Traversing Mechanisms Used on Winding Machines

**The seven mechanisms shown below are used on different types of yarn and coil winding machines. Their fundamentals, however, may be applicable to other machines which require similar changes of motion. Except for the lead screw as used, for example on lathes, these seven represent the operating principles of all well-known, mechanical types of traversing devices.**

E. R. Swanson

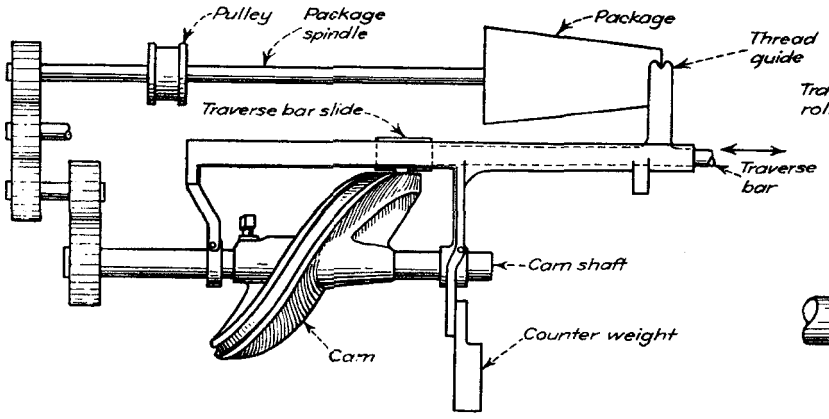



FIG. 1. Package is mounted on belt driven shaft on this precision type winding mechanism. Cam shaft imparts reciprocating motion to traverse bar by means of cam roll that runs in cam groove. Gears determine speed ratio between cam and package. Thread guide is attached to traverse bar. Counterweight keeps thread guide against package.

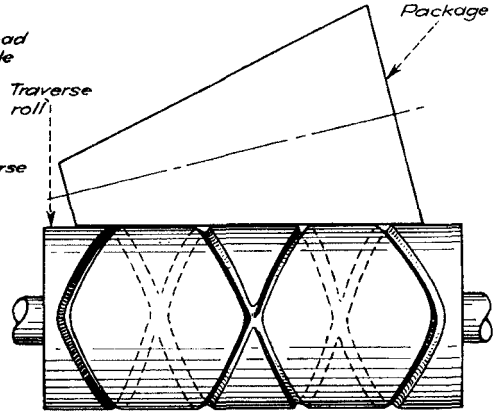



FIG. 2. Package is friction-driven from traverse roll. Yarn is drawn from the supply source by traverse roll and is transferred to package from the continuous groove in the roll. Different winds are obtained by varying the grooved path.

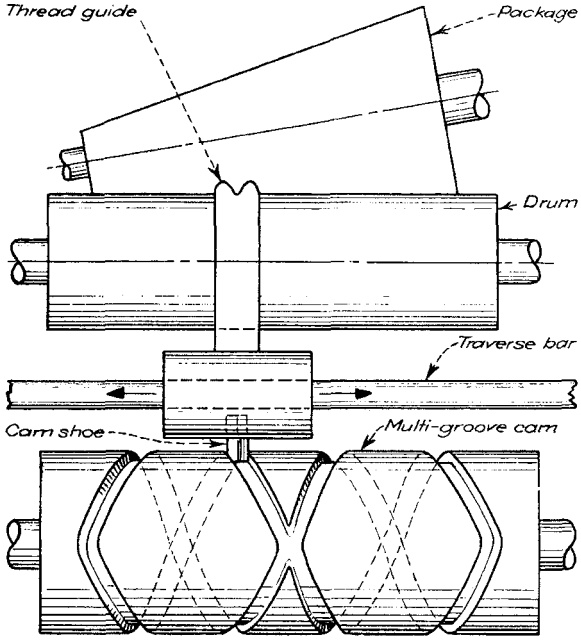



FIG. 4. Drum drives package by friction. Pointed cam shoe which pivots in the bottom side of the thread guide assembly rides in cam grooves and produces reciprocating motion of the thread guide assembly on the traverse bar. Plastic cams have proved quite satisfactory even with fast traverse speeds. Interchangeable cams permit a wide variety of winds.

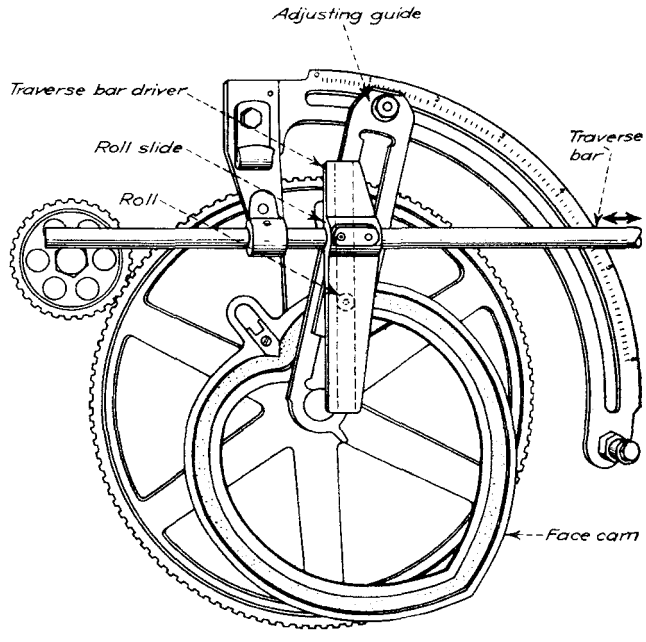



FIG. 5. Roll that rides in heart-shaped cam groove engages slot in traverse bar driver which is attached to the traverse bar. Maximum traverse is obtained when adjusting guide is perpendicular to the driver. As angle between guide and driver is decreased, traverse decreases proportionately. Inertia effects limit this type mechanism to slow speeds.

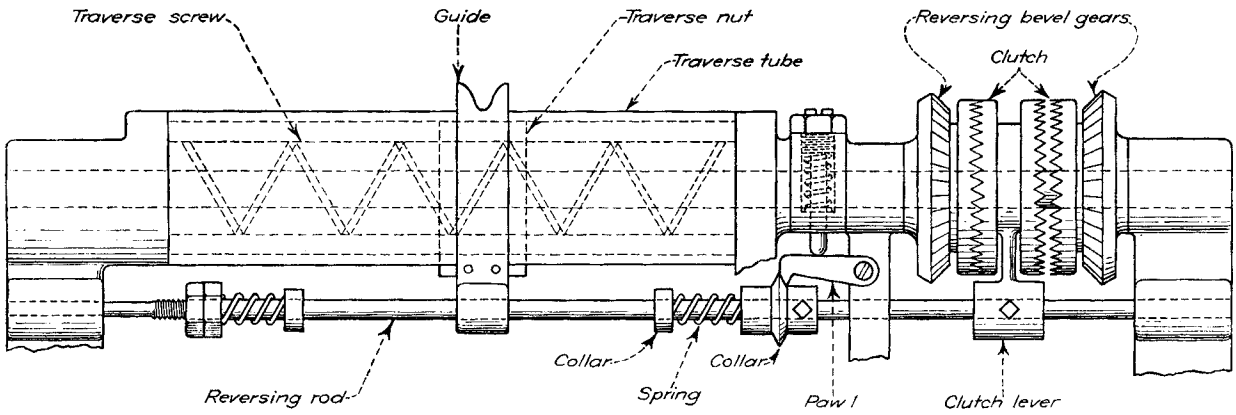



FIG. 3. Reversing bevel gears which are driven by a common bevel gear, drive the shaft carrying the traverse screw. Traverse nut mates with this screw and is connected to the yarn guide. Guide slides along the reversing rod. When nut reaches end of its travel, the thread guide compresses the spring that actuates the pawl and the reversing lever. This engages the clutch that rotates the traverse screw in the opposite direction. As indicated by the large pitch on the screw, this mechanism is limited to low speeds, but permits longer lengths of traverse than most of the others shown.

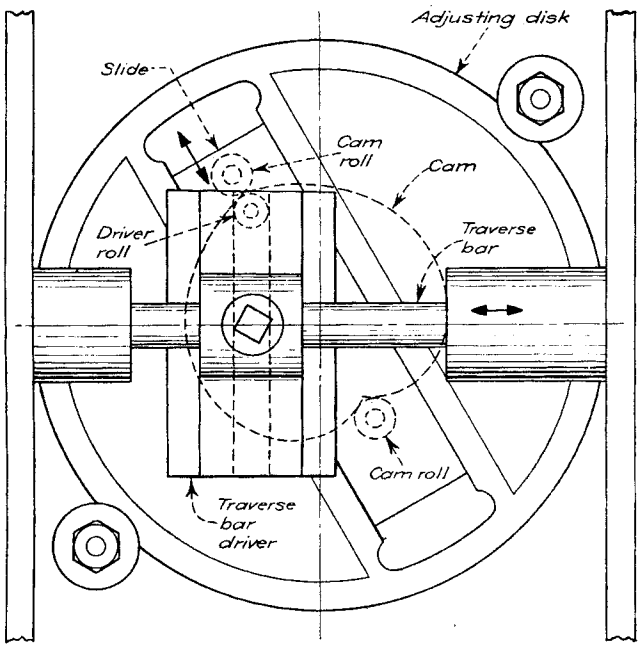



FIG. 6. Two cam rolls that engage heart shaped cam are attached to the slide. Slide has a driver roll that engages a slot in the traverse bar driver. Maximum traverse (to capacity of cam) occurs when adjusting disk is set so slide is parallel to traverse bar. As angle between traverse bar and slide increases, traverse decreases. At 90 deg traverse is zero.

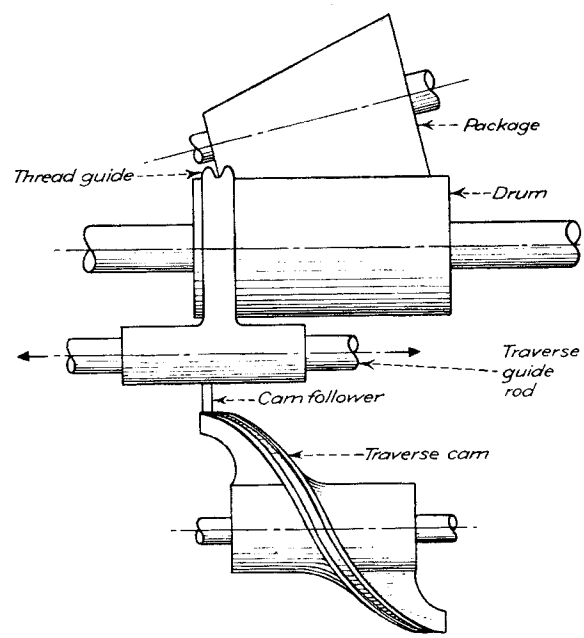



FIG. 7. Traverse cam imparts reciprocating motion to cam follower which drives thread guides on traverse guide rods. Package is friction driven from drum. Yarn is drawn from the supply source through thread guide and transferred to the drum-driven package. Speed of this type of mechanism is determined by the weight of the reciprocating parts.

# Mechanisms for Providing Intermittent Rotary Motion

**Eleven ways of converting uniform angular motion to intermittent angular motion and an explanation of two indirect ways to get this conversion.**

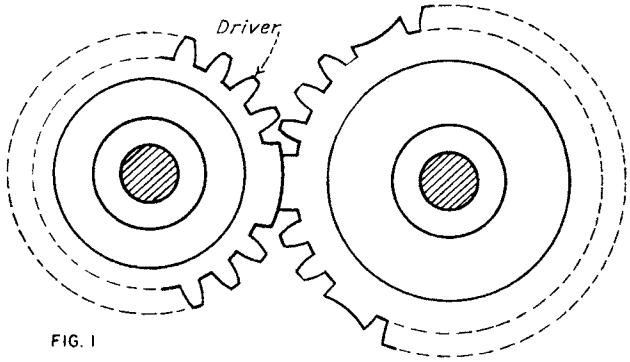



FIG. 1

EXTERNAL TIMING GEARS

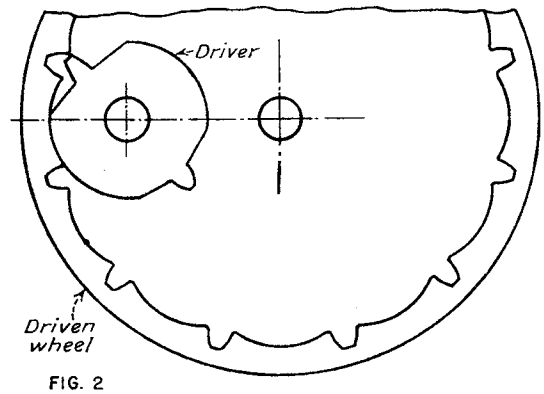



FIG. 2

INTERNAL TIMING GEARS.

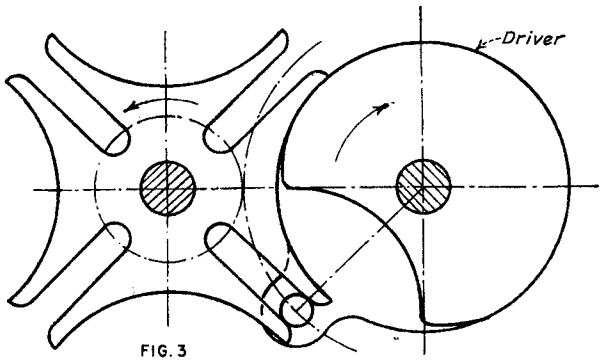



FIG. 3

EXTERNAL GENEVA MECHANISM. Operation is smoother than timing gears. Practical number of slots is from 3 to 18. Duration of dwell is more than 180 deg of driver rotation.

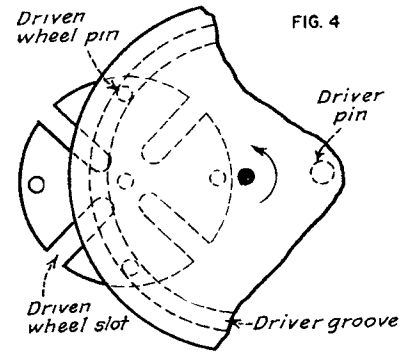



FIG. 4

EXTERNAL GENEVA MECHANISM. Difference from mechanism of of Fig. 3 lies in method of locking. Driver grooves lock driven wheel pins during dwell. During movement, driver pin mates with driven wheel slot.

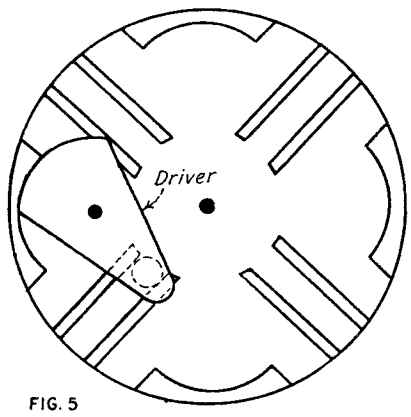



FIG. 5

INTERNAL GENEVA MECHANISM. Driver and driven wheel rotate in same direction. Duration of dwell is more than 180 deg of driver rotation.

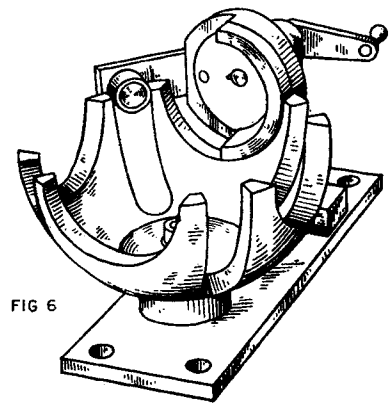
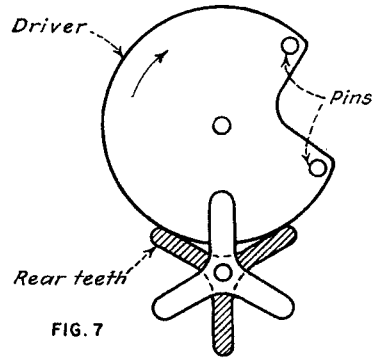
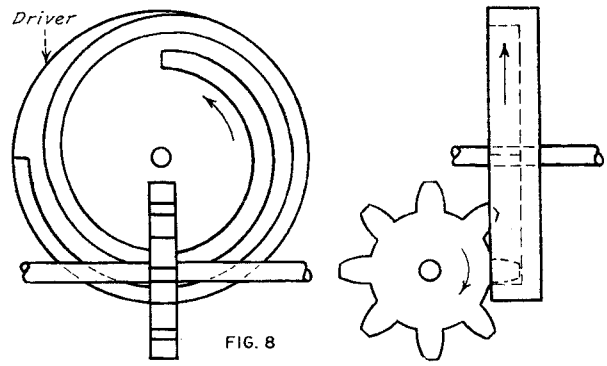


FIG 6

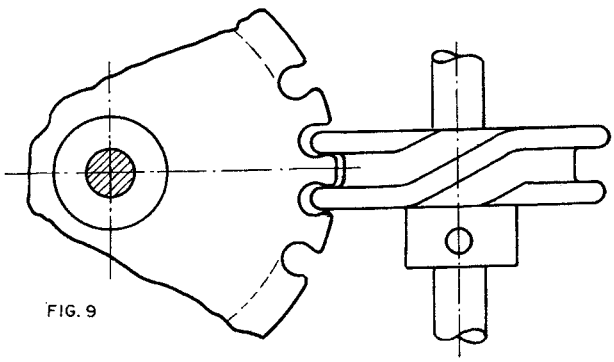
SPHERICAL GENEVA MECHANISM. Driver and driven wheel are on perpendicular shafts. Duration of dwell is exactly 180 deg of driver rotation.



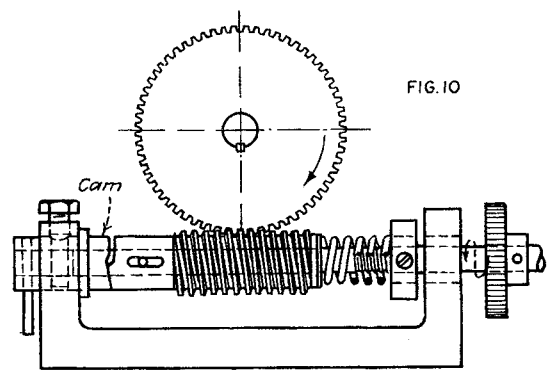

INTERMITTENT COUNTER MECHANISM. One revolution of driver advances driven wheel 120 degrees. Driven wheel rear teeth locked on cam surface during dwell.



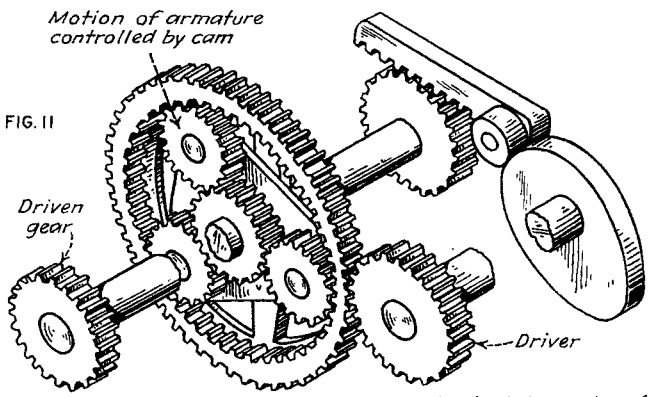

SPIRAL AND WHEEL. One revolution of spiral advances driven wheel 1 tooth. Driven wheel tooth locked in driver groove during dwell.




SPECIAL WORM AND WHEEL. Spiral of Fig. 8 is replaced by special worm.




WORM, CAM AND WHEEL. Standard worm and cam replace special worm of Fig. 9.




SPECIAL PLANETARY GEAR MECHANISM. Principle of relative motion of mating gears illustrated in Fig. 10 can be applied to spur gears in planetary system. Motion of normally fixed planet centers produces intermittent motion of sum gear.

THE CONVERSION of a uniform rotation to an intermittent rotation need not be performed in a single step. Two ways in which this can be carried out in two steps are:

1. Conversion of uniform rotation to reciprocating motion followed by conversion of reciprocating motion to intermittent motion.

2. Conversion of uniform rotation to angular oscillation followed by conversion of angular oscillation to intermittent rotation.

*Fig. 11—Courtesy G. J. Tarbourdet, ASME paper No. 48-SA-18.*

# Intermittent Movements and Mechanisms

SEVERAL RECENTLY PATENTED mechanisms for producing intermittent motion are included in this group of devices. Many of the simpler and well-known mechanisms such as the common ratchet and pawl with its many modifications, escapements, numerous variations of the Geneva mechanisms and timed electrical contactors have been intentionally omitted in favor of the more ingenious combinations of mechanisms for accomplishing intermittent motion under unusual conditions.

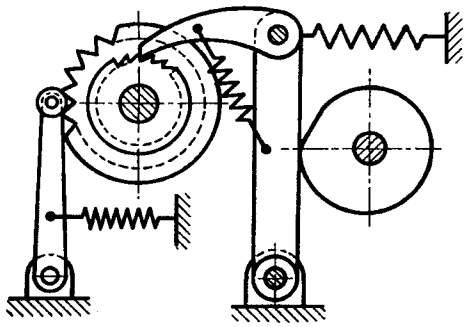


FIG. 1- CAM DRIVEN RATCHET

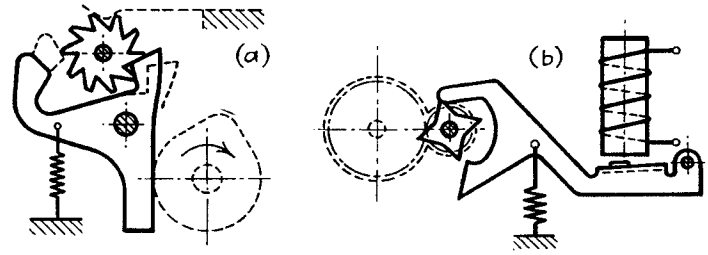



FIG. 3- (a) CAM OPERATED ESCAPEMENT ON A TAXIMETER
(b) SOLENOID OPERATED ESCAPEMENT

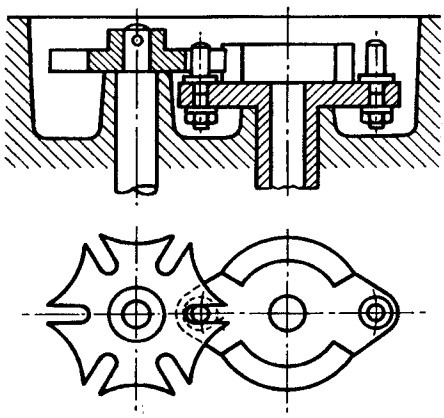


FIG. 2- SIX-SIDED MALTESE CROSS AND DOUBLE DRIVER GIVES 3:1 RATIO

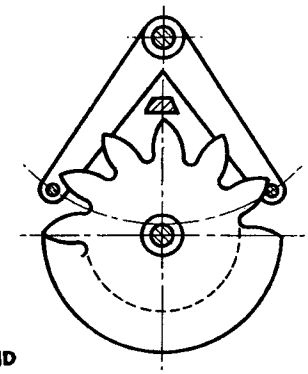


FIG. 4- ESCAPEMENT USED ON AN ELECTRIC METER

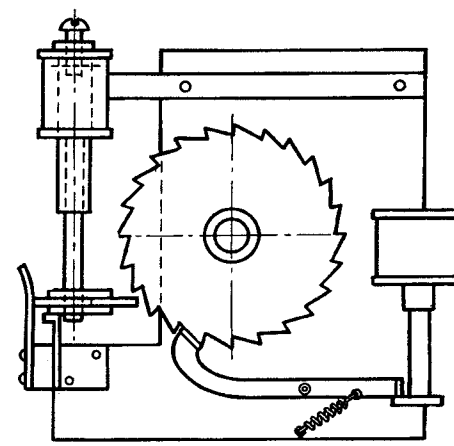


FIG. 5- SOLENOID-OPERATED RATCHET WITH SOLENOID RESETING MECHANISM. A SLIDING WASHER ENGAGES TEETH

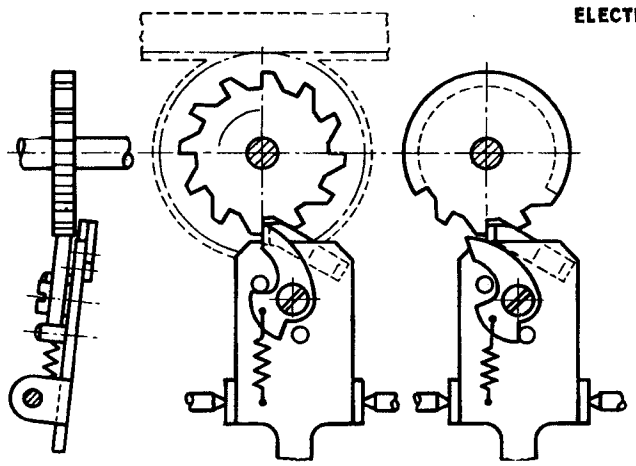


FIG. 6- PLATE OSCILLATING ACROSS PLANE OF RATCHET GEAR ESCAPEMENT CARRIES STATIONARY AND SPRING HELD PAWLS

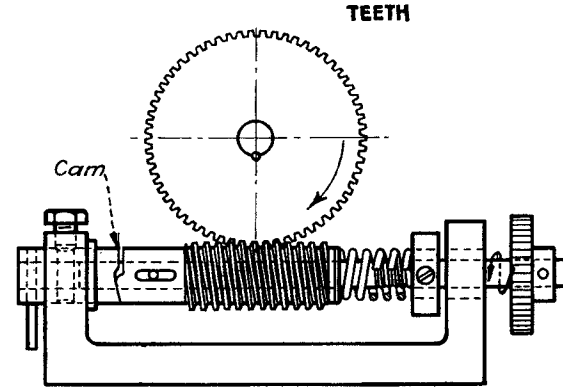



FIG. 7- WORM DRIVE COMPENSATED BY CAM ON WORK SHAFT, PRODUCES INTERMITTENT MOTION OF GEAR

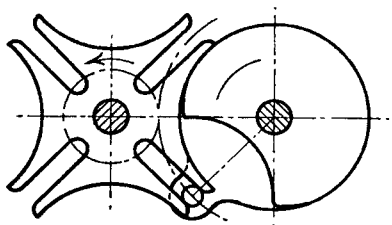


FIG. 8- GENEVA MECHANISM

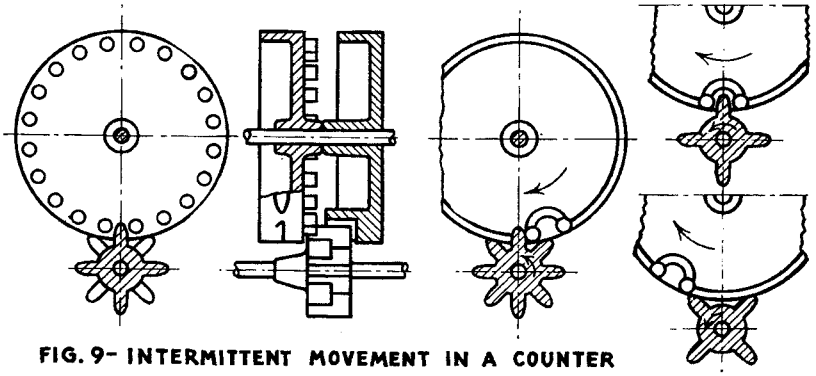
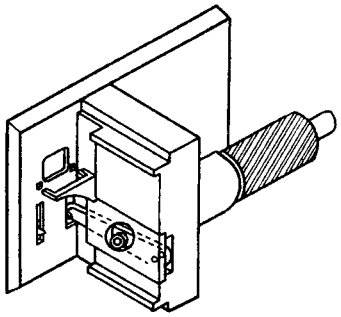
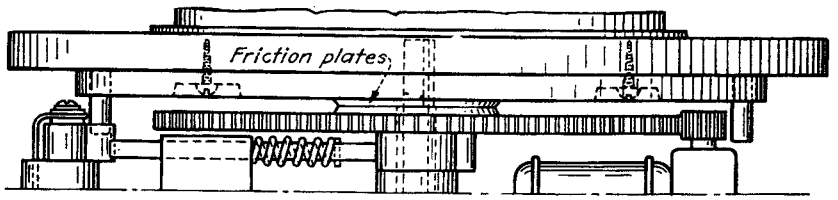


FIG. 9- INTERMITTENT MOVEMENT IN A COUNTER





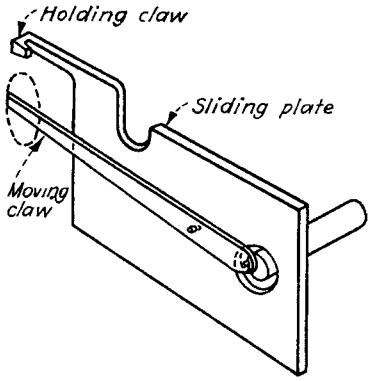




FIG. 10- MOTION PICTURE FILM MOVEMENTS FEATURING SIMPLICITY, FEW OPERATING PARTS, QUIETNESS

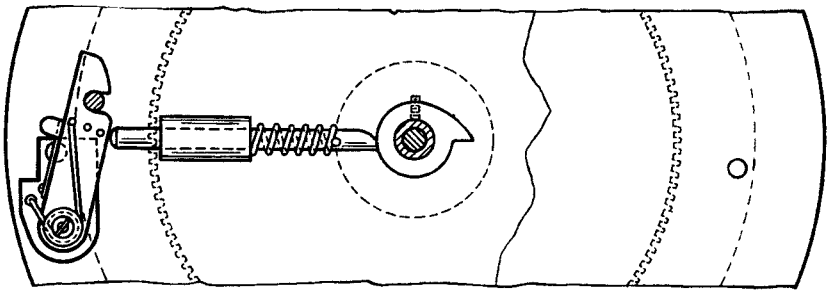


FIG. 11- DISPLAY SIGN MOVEMENT DEPENDS ON FRICTION AND USES A CAM-OPERATED LOCKING MECHANISM. ELECTRICALLY-OPERATED LOCKING MECHANISM HAS ALSO BEEN WORKED OUT BY MEANS OF TIMED-ELECTRIC CONTACTS AND MOTOR-DRIVEN LOCK PARTS

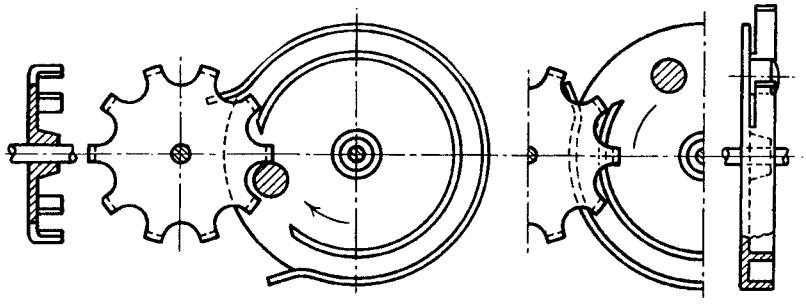


FIG. 12- MODIFICATION OF THE GENEVA MECHANISM

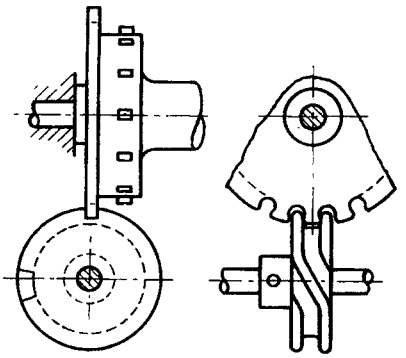


FIG. 13- SIMPLE WORM GEAR MECHANISM

# Friction Devices for Intermittent Rotary Motion

W. M. Halliday

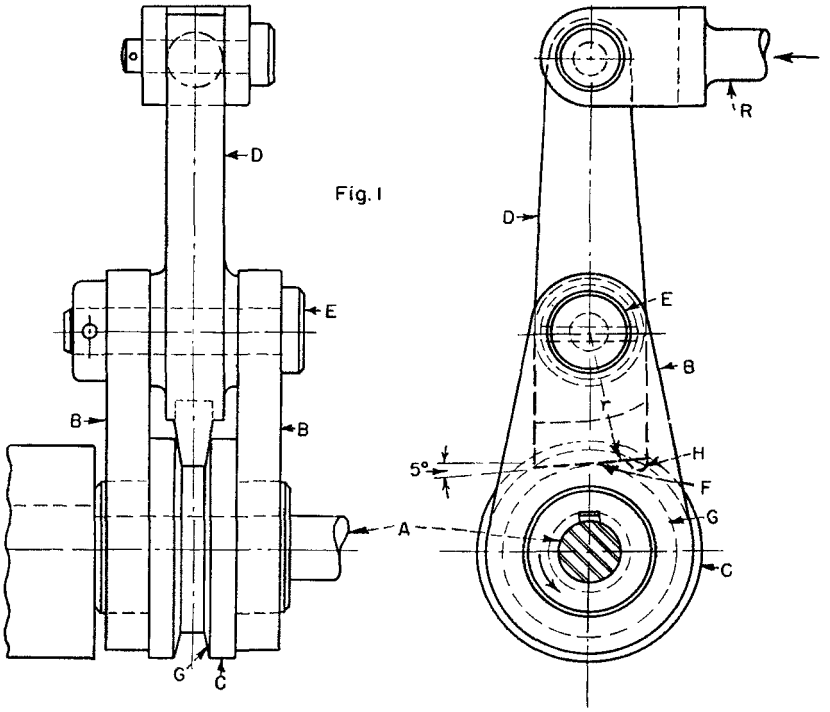



Fig. 1

FIG. 1—WEDGE AND DISK. Consists of shaft *A* supported in bearing block *J*; ring *C* keyed to *A* and containing an annular groove *G*; body *B* which can pivot around the shoulders of *C*; lever *D* which can pivot about *E*; and connecting rod *R* driven by an eccentic (not shown). Lever *D* is rotated counterclockwise about *E* by the connecting rod moving to the left until surface *F* wedges into groove *G*. Continued rotation of *D* causes *A*, *B* and *D* to rotate counterclockwise as a unit about *A*. Reversal of input motion instantly swivels *F* out of *G*, thus unlocking the shaft which remains stationary during return stroke because of friction induced by its load. As *D* continues to rotate clockwise about *E*, node *H*, which is hardened and polished to reduce friction, bears against bottom of *G* to restrain further swiveling. Lever *D* now rotates with *B* around *A* until end of stroke.

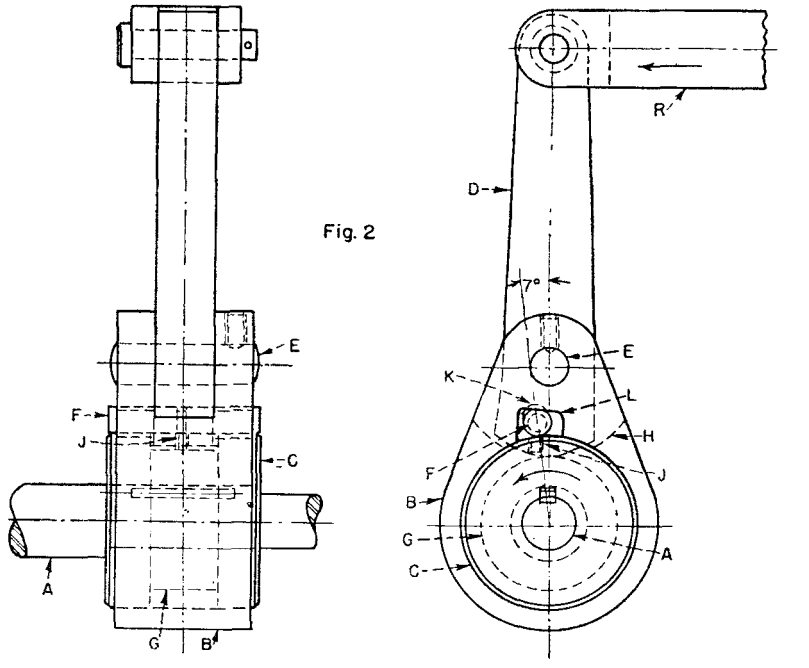



Fig. 2

FIG. 2—PIN AND DISK. Lever *D*, which pivots around *E*, contains pin *F* in an elongated hole *K* which permits slight vertical movement of pin but prevents horizontal movement by means of set screw *J*. Body *B* can rotate freely about shaft *A* and has cut-outs *L* and *H* to allow clearances for pin *F* and lever *D*, respectively. Ring *C*, which is keyed to shaft *A*, has an annular groove *G* for clearance of the tip of lever *D*. Counterclockwise motion of lever *D* actuated by the connecting rod jams pin between *C* and the top of cut-out *L*. This occurs about seven degrees from the vertical axis. *A*, *B* and *D* are now locked together and rotate about *A*. Return stroke of *R* pivots *D* around *E* clockwise and unwedges pin until it strikes side of *L*. Continued motion of *R* to the right rotates *B* and *D* clockwise around *A* while the uncoupled shaft remains stationary because of its load.

Friction devices are free from the common disadvantages inherent in conventional pawl and ratchet drives such as: (1) noisy operation; (2) backlash needed for engagement of pawl; (3) load concentrated on one tooth of the ratchet; and (4) pawl engagement dependent on an external spring.

The five mechanisms presented here convert the reciprocating motion of a connecting rod into intermittent rotary motion. The connecting rod stroke to the left drives a shaft counterclockwise; shaft is uncoupled and remains stationary during the return stroke of connecting rod to the right.

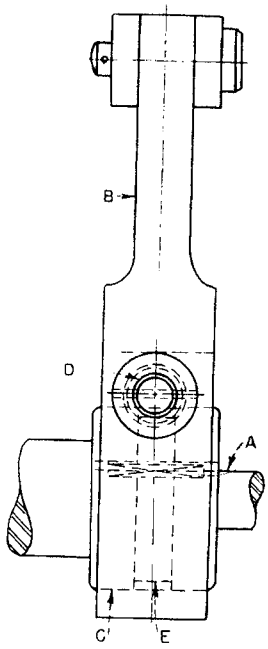



Fig. 3

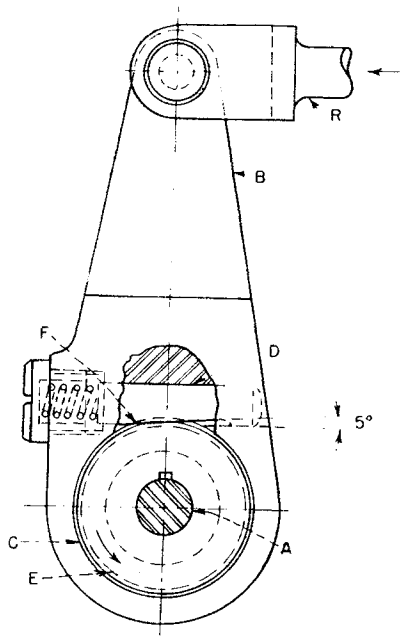



FIG. 3—SLIDING PIN AND DISK. Counterclockwise movement of body *B* about shaft *A* draws pin *D* to the right with respect to body *B*, aided by spring pressure, until the flat bottom *F* of pin is wedged against annular groove *E* of ring *C*. Bottom of pin is inclined about five degrees for optimum wedging action. Ring *C* is keyed to *A* and parts *A*, *C*, *D* and *B* now rotate counterclockwise as a unit until end of connecting rod's stroke. Reversal of *B* draws pin out of engagement so that *A* remains stationary while body completes its clockwise rotation.

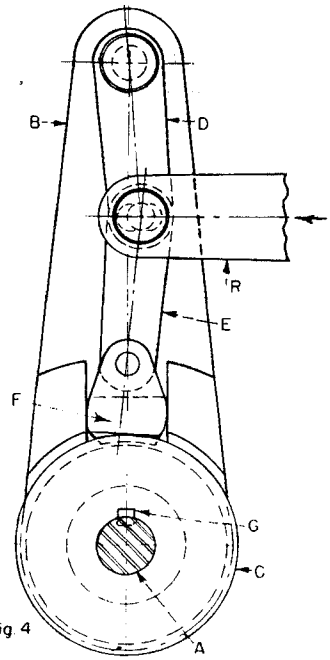



Fig. 4

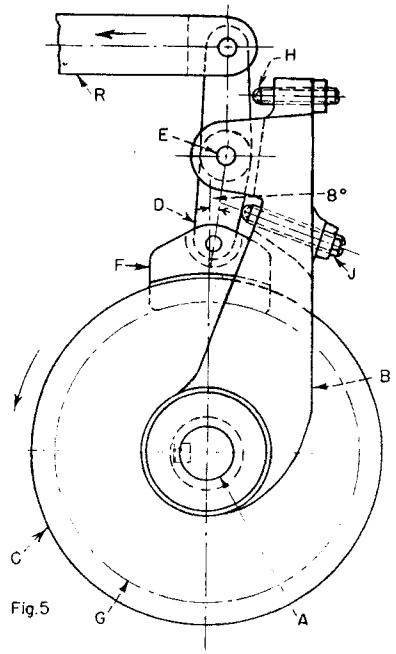



Fig. 5

FIG. 4—TOGGLE LINK AND DISK. Input stroke of connecting rod *R* to the left wedges block *F* in groove *G* by straightening toggle links *D* and *E*. Body *B*, toggle links and ring *C* which is keyed to shaft *A*, rotate counterclockwise together about *A* until end of stroke. Reversal of connecting rod motion lifts block, thus uncoupling shaft, while body *B* continues clockwise rotation until end of stroke.

FIG. 5—ROCKER ARM AND DISK. Lever *D*, activated by the reciprocating bar *R* moving to the left, rotates counterclockwise on pivot *E* thus wedging block *F* into groove *G* of disk *C*. Shaft *A* is keyed to *C* and rotates counterclockwise as a unit with body *B* and lever *D*. Return stroke of *R* to the right pivots *D* clockwise about *E* and withdraws block from groove so that shaft is uncoupled while *D*, striking adjusting screw *H*, travels with *B* about *A* until completion of stroke. Adjusting screw *J* prevents *F* from jamming in groove.

# Basic Types of Variable Speed Friction Drives

Haim Murro

THESE DRIVES are used to transmit both high torque, as on power spindle presses, and low torque, as in laboratory instruments. All perform best if used to reduce and not to increase speed. All friction drives have a certain degree of slip due to imperfect rolling of the friction members, but by correct design this slip can be held constant, resulting in constant speed of the driven member. Variations in load should be compensated by inertia masses on the driven end. Springs or similar elastic members can be used to keep the friction parts in constant contact and exert the force necessary to create the friction. In some cases, gravity will take place of such members. Specially made friction materials are generally recommended, but leather or rubber are often satisfactory. Normally only one of the friction members is made or lined with this material, while the other is metal.

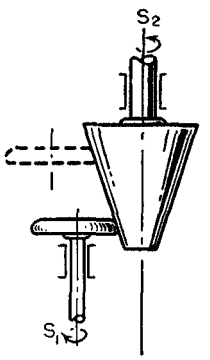



Fig. 1

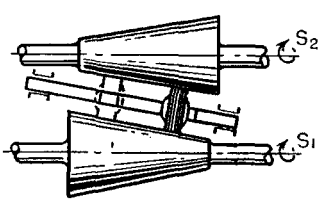



Fig. 2

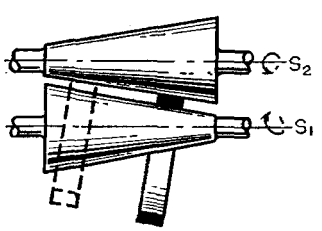



Fig. 3

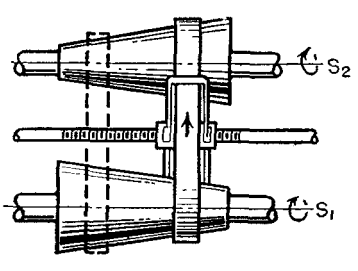



Fig. 4

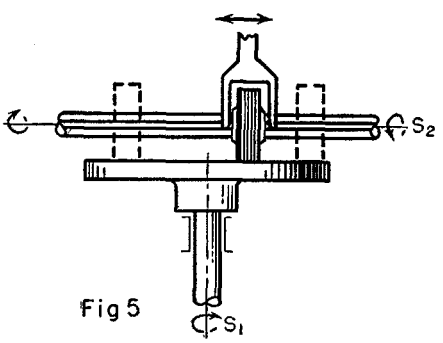



Fig 5

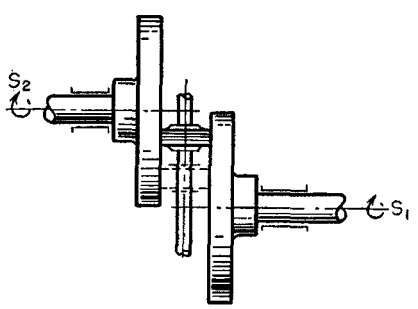



Fig.6

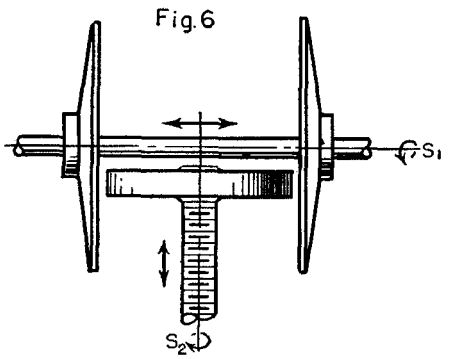



Fig.7

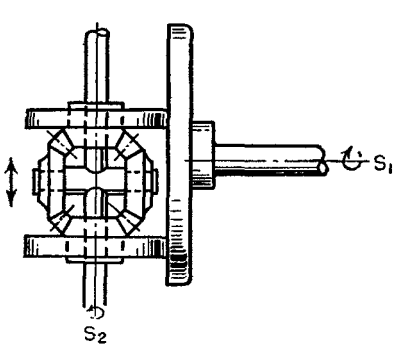



Fig.8

**Fig. 1—Cone and roller drive. Speed varies in accordance with axial position of roller on the cone and its operating diameter.**

**Fig. 2—Two cones and a free spinning roller that can be moved along the surfaces of the cones.**

**Fig. 3—Two cones and a free endless leather belt between them. The belt is shifted between cone faces to obtain the desired speed ratio.**

**Fig. 4—Two cones belted by a flat movable belt. Two metal fingers position belt along faces of the cones.**

**Fig. 5—A disk and roller drive. Roller is moved radially on the disk. Speed ratio depends upon the operating diameter of disk. Direction of relative rotation of shafts is reversed when roller is moved past the center of the disk as indicated by dotted lines.**

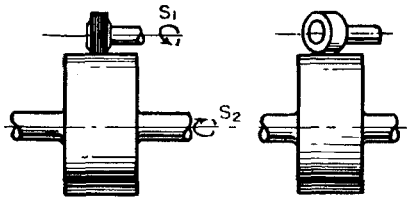



Fig. 9

Fig. 6—Two disks and a free spinning roller, movable between them. Change of speed will be fast, because the operating diameters of the disks change in an inverse ratio.

Fig. 7—Two disks mounted on same shaft and a roller mounted on a threaded spindle. Contact of roller can be changed from one disk to the other to change direction of rotation. Rotation will be accelerated or decelerated with movement of the screw.

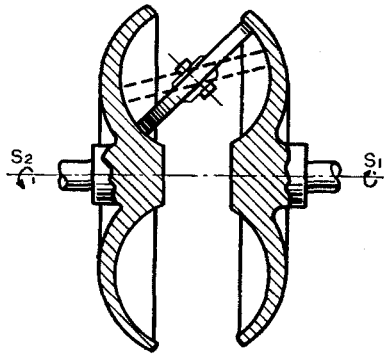



Fig. 12

Fig. 8—Disk and two differential rollers. Rollers and their bevel gears are free to rotate on shaft, $S_2$. Other two bevel gears are free to rotate on pins connected by $S_2$. Good for high reduction and accurate adjustment of speed. $S_2$ will have the differential speed of the two rollers. Differential assembly is movable across face of disk.

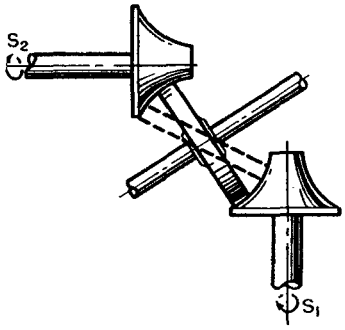



Fig. 10

Fig. 9—Drum and roller. Change of speed is effected by skewing the roller relative to the drum.

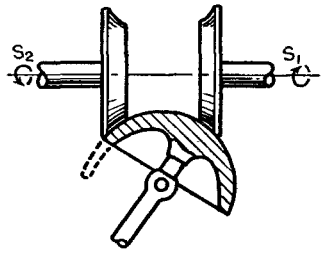



Fig. 13

Fig. 10—Two spherical cones on intersecting shafts and a free roller.

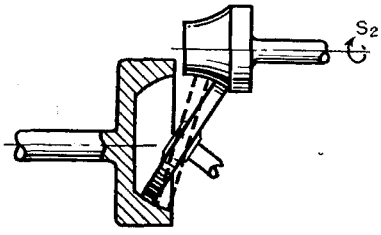



Fig. 11

Fig. 11—Spherical cone and groove with a roller. Suitable when small adjustments in speed are desired.

Fig. 12—Two disks with torus contours and a free rotating roller.

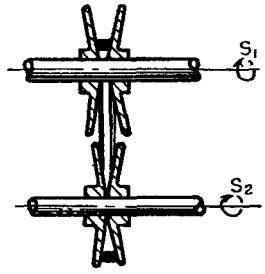



Fig. 14

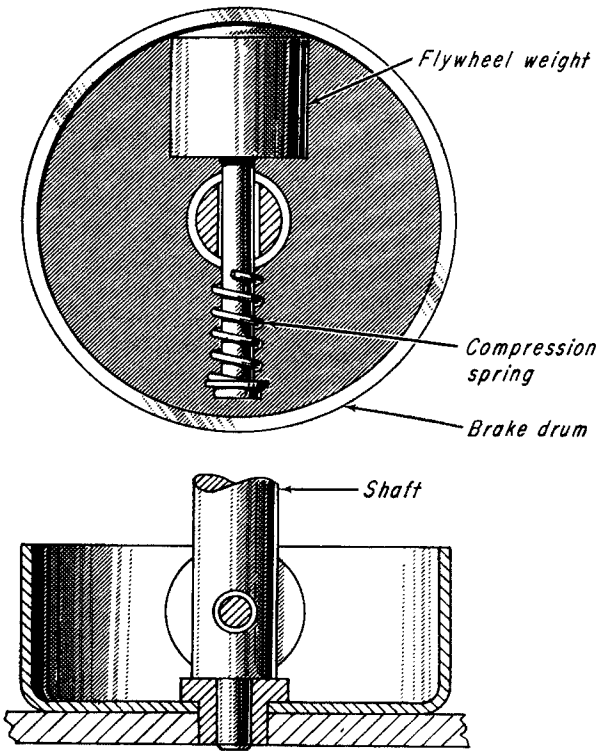
Fig. 13—Two disks with a spherical free rotating roller.

Fig. 14—Split pulleys for V belts. Effective diameter of belt grip can be adjusted by controlling distance between two parts of pulley.

# Speed Control for Small Mechanisms

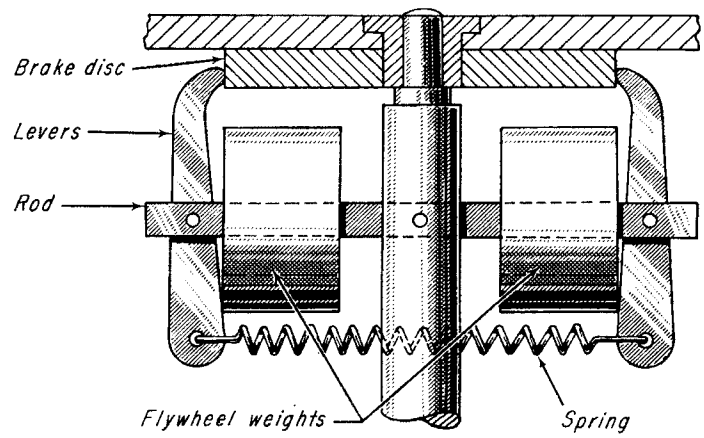
**Friction devices, actuated by centrifugal force, automatically keep speed constant regardless of variation of load or driving force.**

Federico Strasser



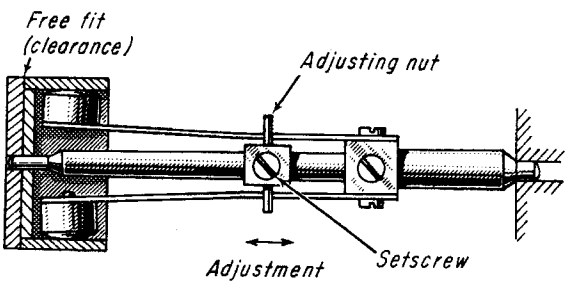

1

**WEIGHT** counterbalanced by a spring brakes the shaft when rotation speed becomes too high. Braking area is small.



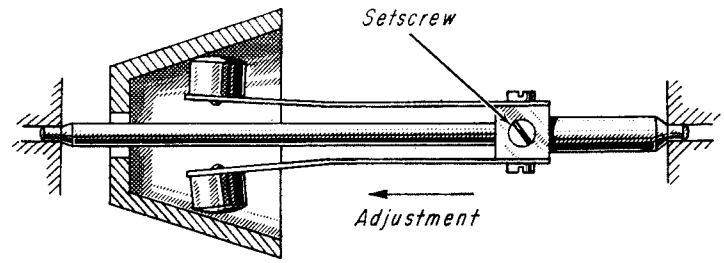

2

**WEIGHT-ACTUATED LEVERS** make this arrangement suitable where high braking moments are required.



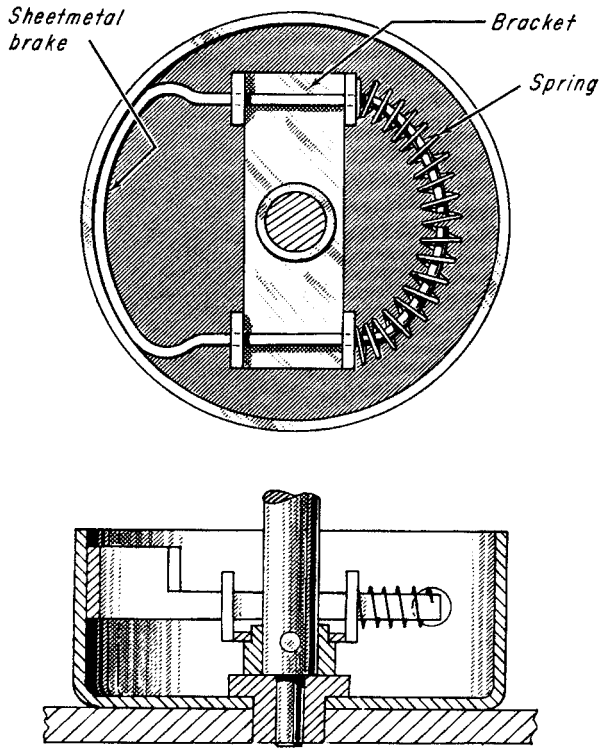

5

**ADJUSTMENT** of speed at which this device starts to brake is quick and easy. Adjusting nut is locked in place with setscrew.

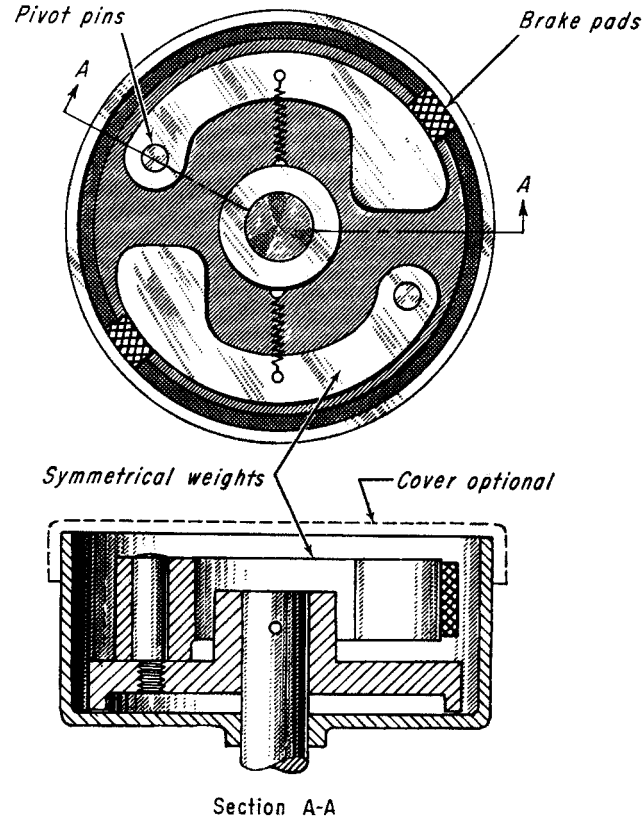



6

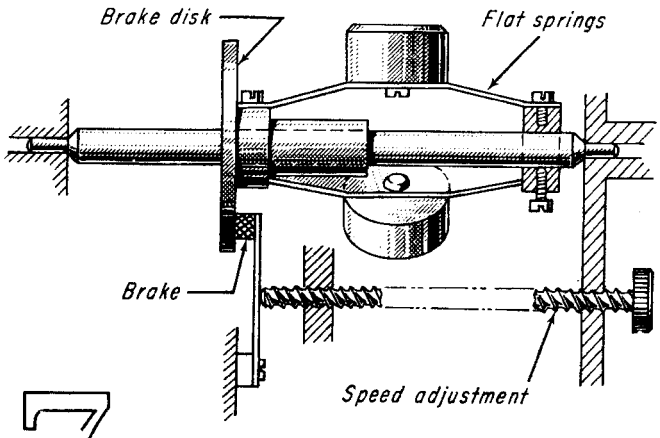
**TAPERED BRAKE DRUM** is another way of providing for varying speed-control. The adjustment is again locked.



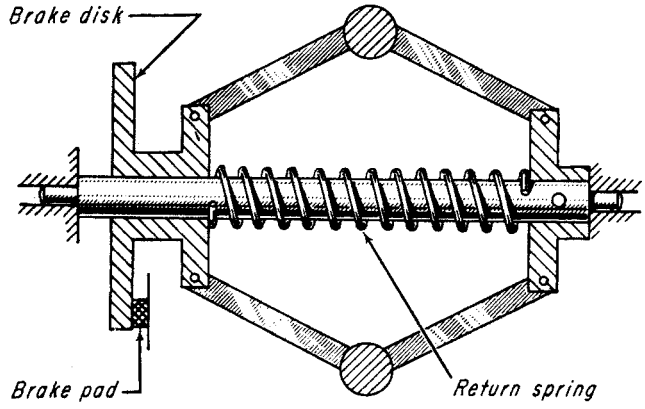

3 **SHEETMETAL BRAKE** provides larger braking area than previous design. Operation is thus more even and cooler.




4 **SYMMETRICAL WEIGHTS** give even braking action when they pivot outward. Entire action can be enclosed.




7 **THREE FLAT SPRINGS** carry weights that provide brake force upon rotation. Device can be provided with adjustment.




8 **TYPICAL GOVERNOR** action of swinging weights is utilized here. As in the previous device, adjustment is optional.

# Mechanical Devices for Automatically Governing Speed

**Speed governors, designed to maintain speeds of machines within reasonably constant limits irrespective of loads, may depend for their action upon centrifugal force or cam linkages. Other types may utilize pressure differentials and fluid velocities as their actuating media. Examples of these governing devices are illustrated.**

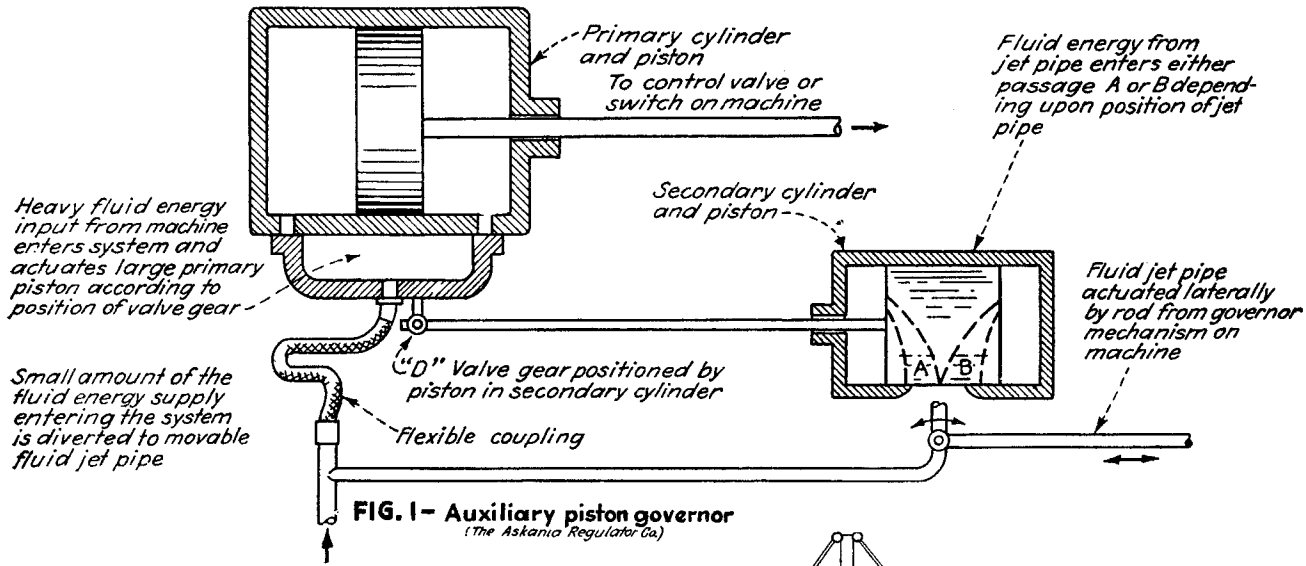

Primary cylinder and piston
To control valve or switch on machine
Fluid energy from jet pipe enters either passage A or B depending upon position of jet pipe
Secondary cylinder and piston
Heavy fluid energy input from machine enters system and actuates large primary piston according to position of valve gear
Fluid jet pipe actuated laterally by rod from governor mechanism on machine
"D" Valve gear positioned by piston in secondary cylinder
A
B
Small amount of the fluid energy supply entering the system is diverted to movable fluid jet pipe
Flexible coupling

**FIG. 1– Auxiliary piston governor**
*(The Askania Regulator Co.)*

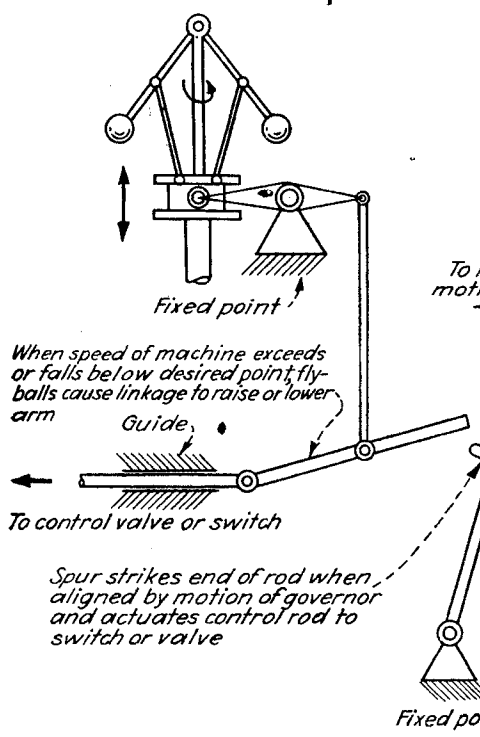

Fixed point
When speed of machine exceeds or falls below desired point, flyballs cause linkage to raise or lower arm
Guide
To control valve or switch
Spur strikes end of rod when aligned by motion of governor and actuates control rod to switch or valve
To reciprocating motion on machine
Fixed point

**FIG. 2– Hit-and-miss governor**

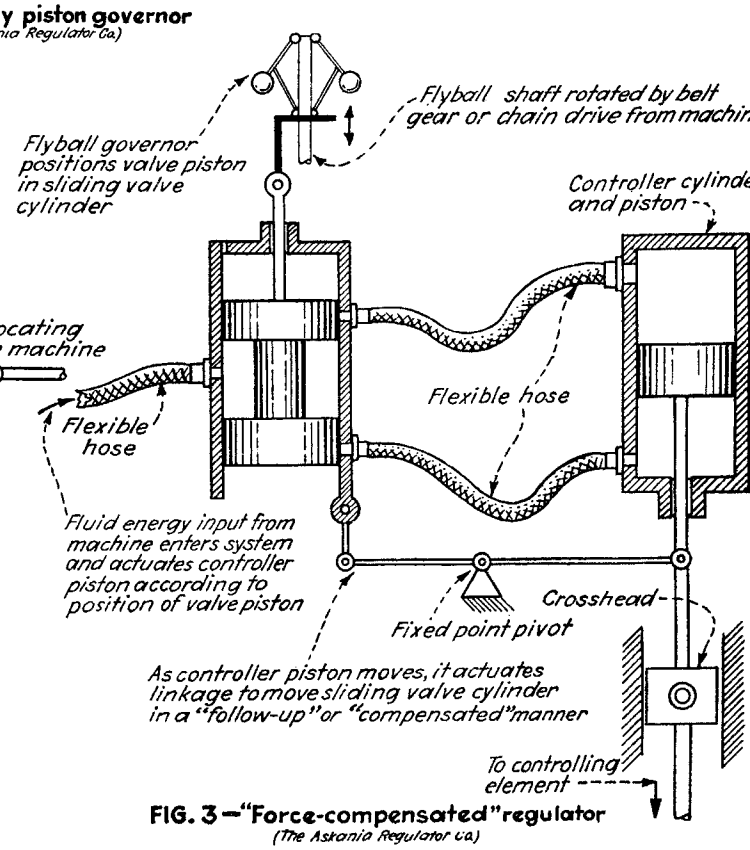

Flyball governor positions valve piston in sliding valve cylinder
Flyball shaft rotated by belt gear or chain drive from machine
Controller cylinder and piston
Flexible hose
Flexible hose
Fluid energy input from machine enters system and actuates controller piston according to position of valve piston
Crosshead
Fixed point pivot
As controller piston moves, it actuates linkage to move sliding valve cylinder in a "follow-up" or "compensated" manner
To controlling element

**FIG. 3–"Force-compensated" regulator**
*(The Askania Regulator Co.)*

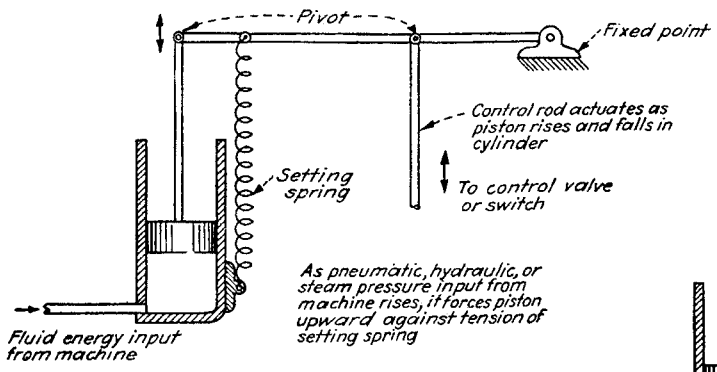

Pivot
Fixed point
Control rod actuates as piston rises and falls in cylinder
Setting spring
To control valve or switch
As pneumatic, hydraulic, or steam pressure input from machine rises, it forces piston upward against tension of setting spring
Fluid energy input from machine

FIG. 4– Pressure-actuated governor

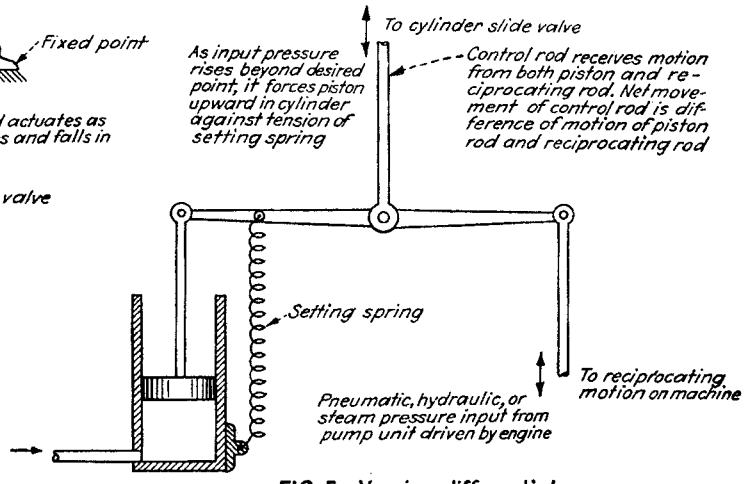

To cylinder slide valve
As input pressure rises beyond desired point, it forces piston upward in cylinder against tension of setting spring
Control rod receives motion from both piston and reciprocating rod. Net movement of control rod is difference of motion of piston rod and reciprocating rod
Setting spring
To reciprocating motion on machine
Pneumatic, hydraulic, or steam pressure input from pump unit driven by engine

FIG. 5– Varying differential governor

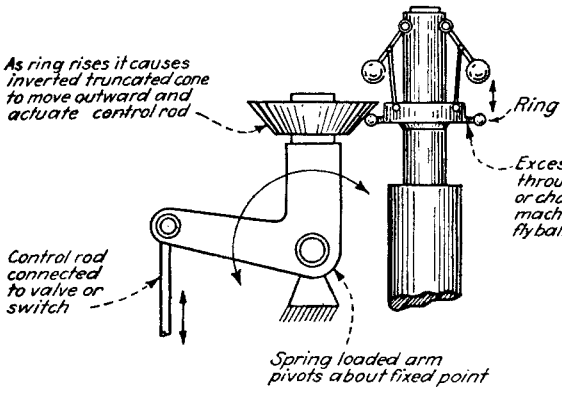

As ring rises it causes inverted truncated cone to move outward and actuate control rod
Ring
Excessive input through belt, shaft, or chain drive from machine causes fly balls to raise ring
Control rod connected to valve or switch
Spring loaded arm pivots about fixed point

FIG. 6– Centrifugal governor

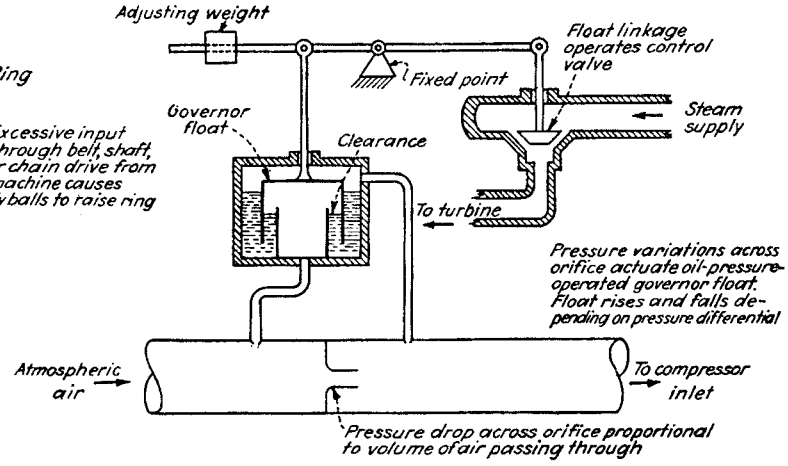

Adjusting weight
Float linkage operates control valve
Fixed point
Governor float
Clearance
Steam supply
To turbine
Pressure variations across orifice actuate oil-pressure-operated governor float. Float rises and falls depending on pressure differential
Atmospheric air
To compressor inlet
Pressure drop across orifice proportional to volume of air passing through

FIG. 7– Constant volume governor

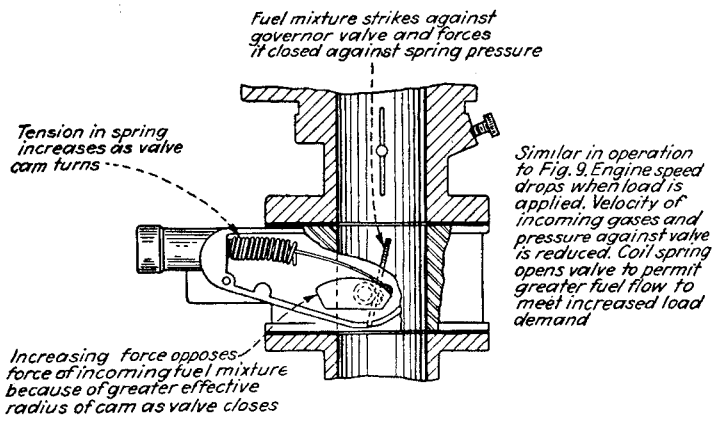

Fuel mixture strikes against governor valve and forces it closed against spring pressure
Tension in spring increases as valve cam turns
Similar in operation to Fig. 9. Engine speed drops when load is applied. Velocity of incoming gases and pressure against valve is reduced. Coil spring opens valve to permit greater fuel flow to meet increased load demand
Increasing force opposes force of incoming fuel mixture because of greater effective radius of cam as valve closes

FIG. 8– Velocity-type governor (coil spring)

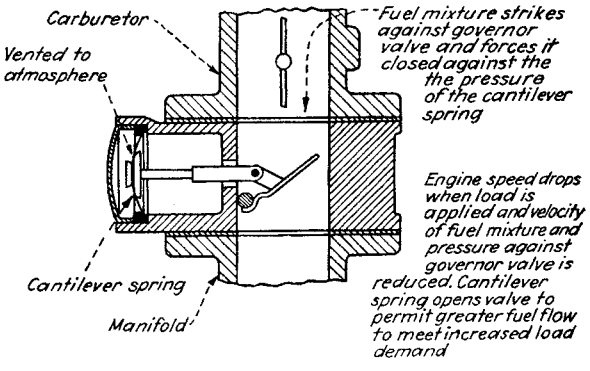

Carburetor
Vented to atmosphere
Fuel mixture strikes against governor valve and forces it closed against the the pressure of the cantilever spring
Engine speed drops when load is applied and velocity of fuel mixture and pressure against governor valve is reduced. Cantilever spring opens valve to permit greater fuel flow to meet increased load demand
Cantilever spring
Manifold

FIG. 9– Velocity-type governor (cantilever spring)

# Accelerated and Decelerated Linear motion Elements

John E. Hyler

**When ordinary rotary cams cannot be conveniently applied, the mechanisms here presented, or adaptations of them, offer a variety of interesting possibilities for obtaining either acceleration or deceleration, or both**

FIG. 1—Slide block moves at constant rate of reciprocating travel and carries both a pin for mounting link *B* and a stud shaft on which the pinion is freely mounted. Pinion carries a crankpin for mounting link *D* and engages stationary rack, the pinion may make one complete revolution at each forward stroke of slide block and another in opposite direction on the return—or any portion of one revolution in any specific instance. Many variations can be obtained by making the connection of link *F* adjustable lengthwise along link that operates it, by making crankpin radially adjustable, or by making both adjustable.

FIG. 2—Drive rod, reciprocating at constant rate, rocks link *BC* about pivot on stationary block and, through effect of toggle, causes decelerative motion of driven link. As drive rod advances toward right, toggle is actuated by encountering abutment and the slotted link *BC* slides on its pivot while turning. This lengthens arm *B* and shortens arm *C* of link *BC*, with decelerative effect on driven link. Toggle is spring-returned on the return stroke and effect on driven link is then accelerative.

FIG. 3—Same direction of travel for both the drive rod and the driven link

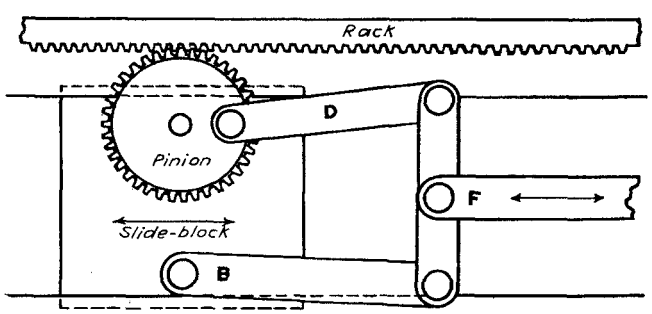



FIG. 1

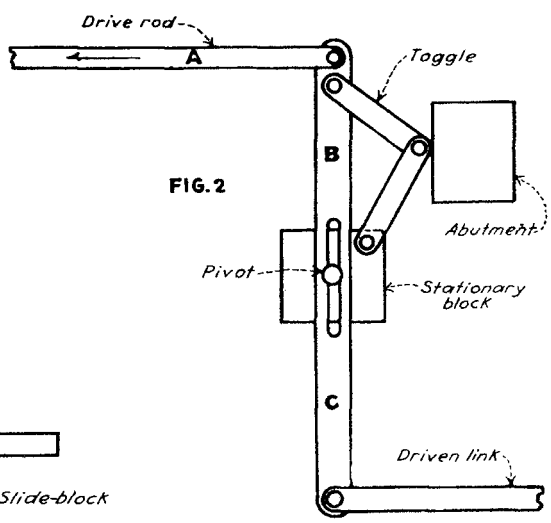



FIG. 2

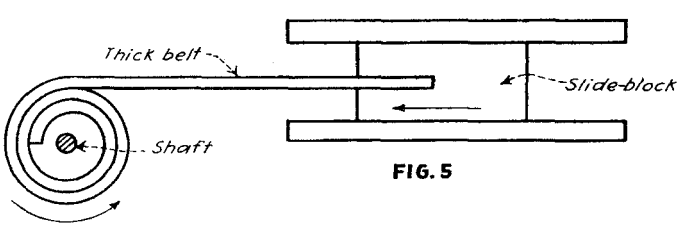



FIG. 5

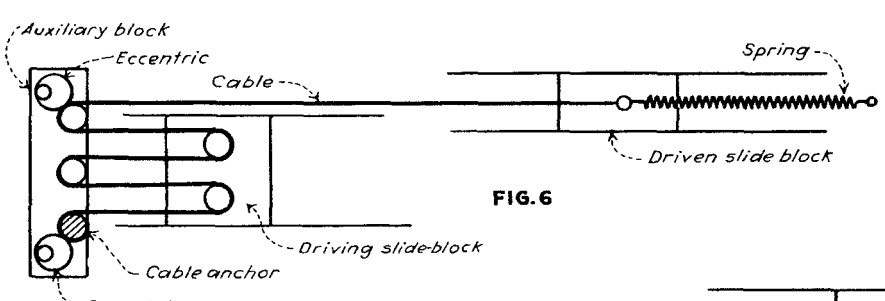



FIG. 6

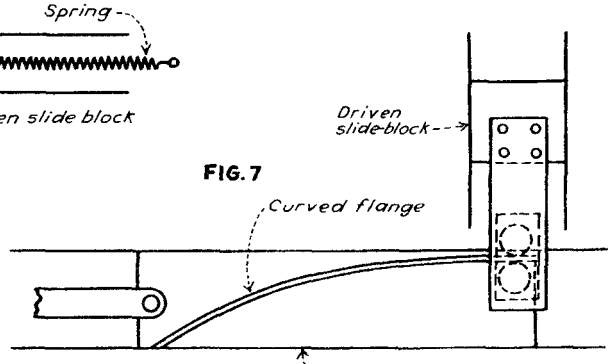



FIG. 7

is provided by this variation of the preceding mechanism. Here, acceleration is in direction of arrows and deceleration occurs on return stroke. Accelerative effect becomes less as toggle flattens.

FIG. 4—Bellcrank motion is accelerated as rollers are spread apart by curved member on end of drive rod, thereby in turn accelerating motion of slide block. Driven elements must be spring-returned to close system.

FIG. 5—Constant-speed shaft winds up thick belt, or similar flexible member, and increase in effective radius causes accelerative motion of slide block. Must be spring- or weight-returned on reversal.

FIG. 6 — Auxiliary block, carrying sheaves for cable which runs between driving and driven slide blocks, is mounted on two synchronized eccentrics. Motion of driven block is equal to length of cable paid out over sheaves resulting from additive motions of the driving and the auxiliary blocks.

FIG. 7—Curved flange on driving slide block is straddled by rollers pivotally mounted in member connected to driven slide block. Flange can be curved to give desired acceleration or deceleration, and mechanism is self-returned.

FIG. 8—Stepped acceleration of the driven slide block is effected as each of the three reciprocating sheaves progressively engages the cable. When the third acceleration step is reached, the driven slide block moves six times faster than the drive rod.

FIG. 9—Form-turned nut, slotted to travel on rider, is propelled by reversing screw shaft, thus moving concave roller up and down to accelerate or decelerate slide block.

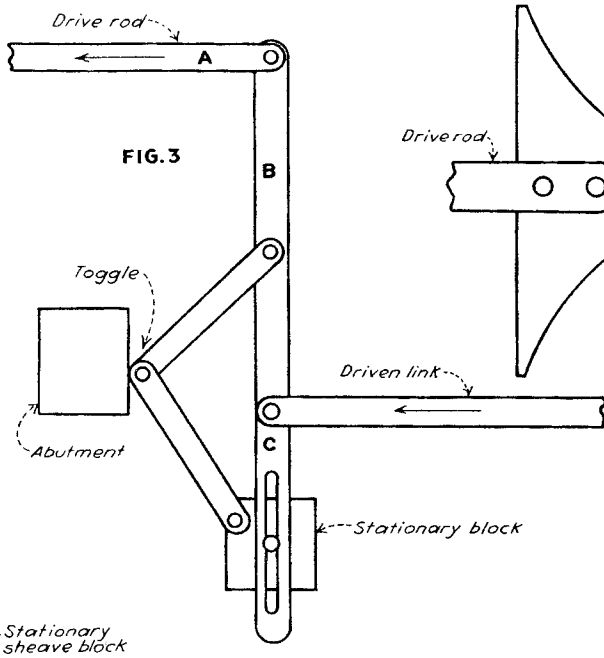



FIG. 3

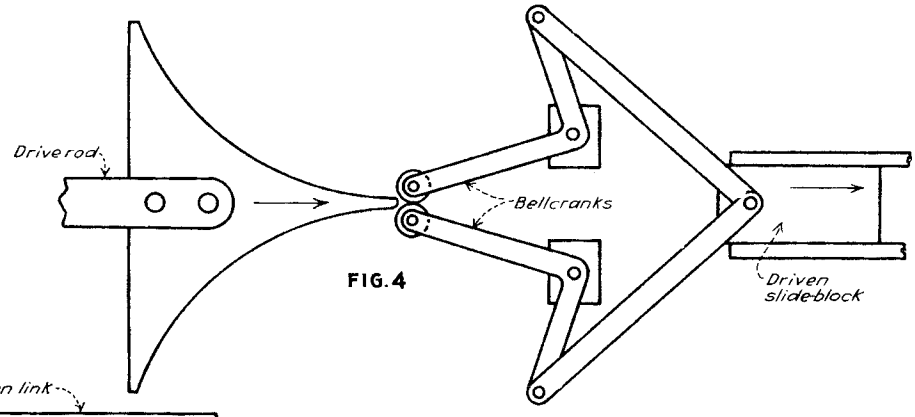



FIG. 4

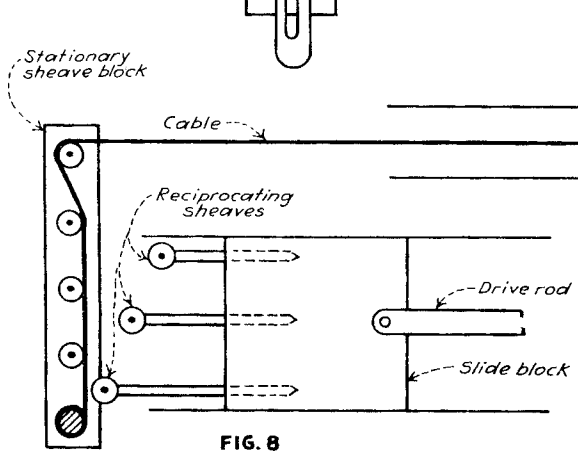



FIG. 8

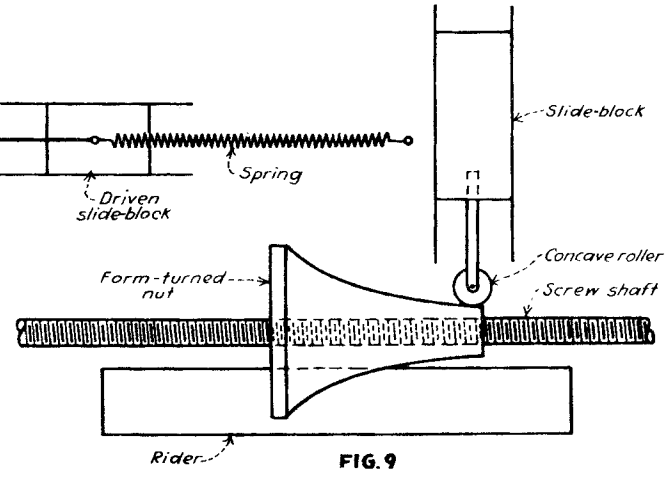
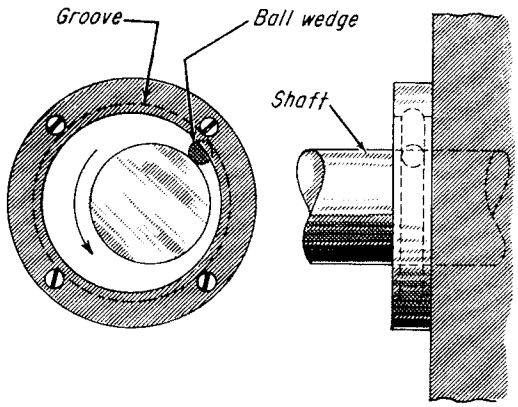



FIG. 9

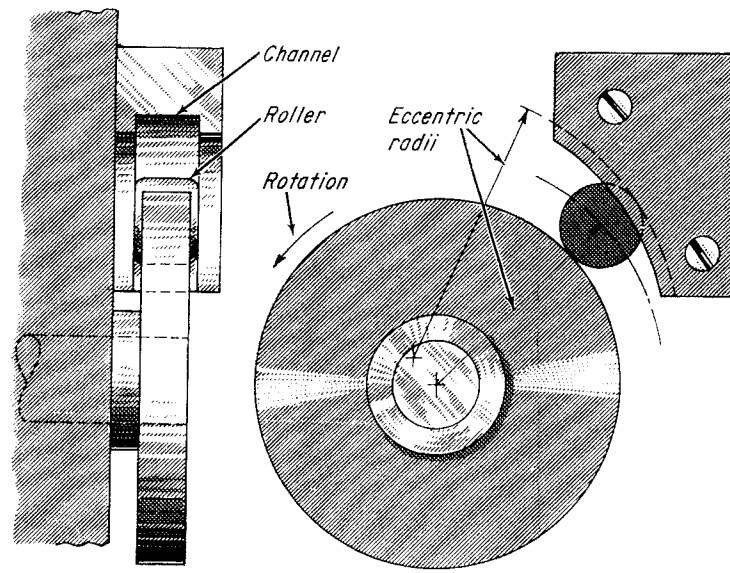
# How to Prevent reverse Rotation

**Wedges, ratchet-and-pawl arrangements, internal pins, friction pads, and sliding key action can protect mechanisms form torque feedback.**

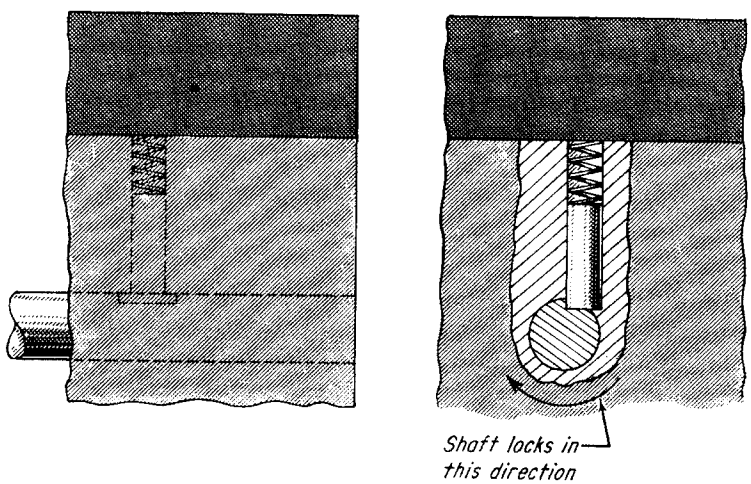
L. Kasper



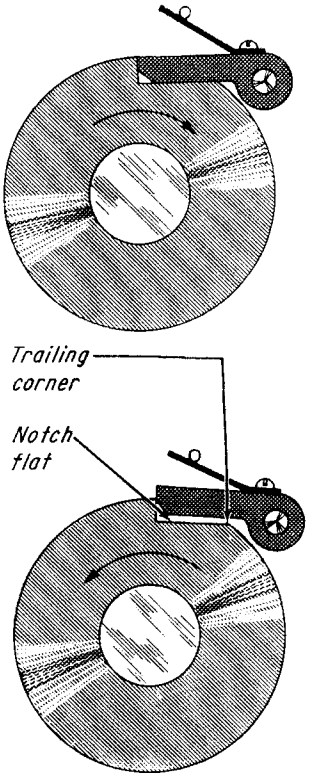

1 **INTERNALLY GROOVED** ring is located eccentrically with the rotating shaft. Ball wedges if shaft reverses direction of rotation.



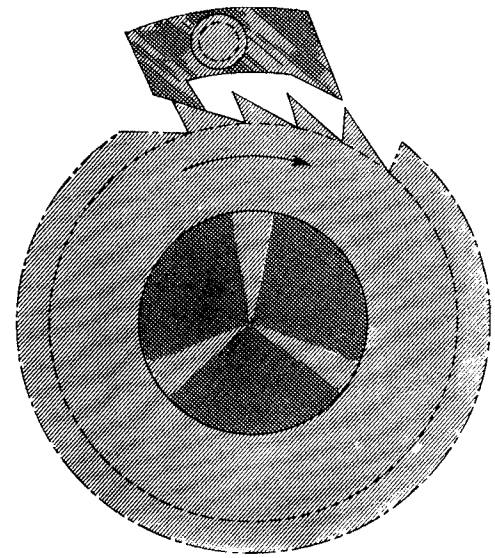

2 **ROLLER WEDGES** when direction of rotation is reversed as in previous device. For vertical shafts, hold roller in place with a spring.



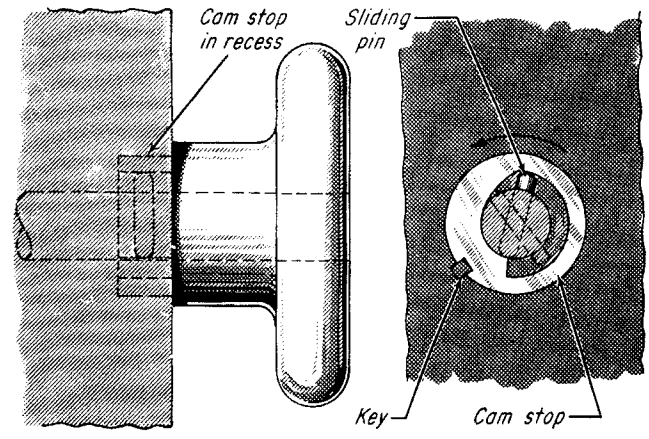

5 **SPRING-LOADED PIN** device is applicable when no part of the shaft is exposed. Blacklash here can be equal to almost one revolution.



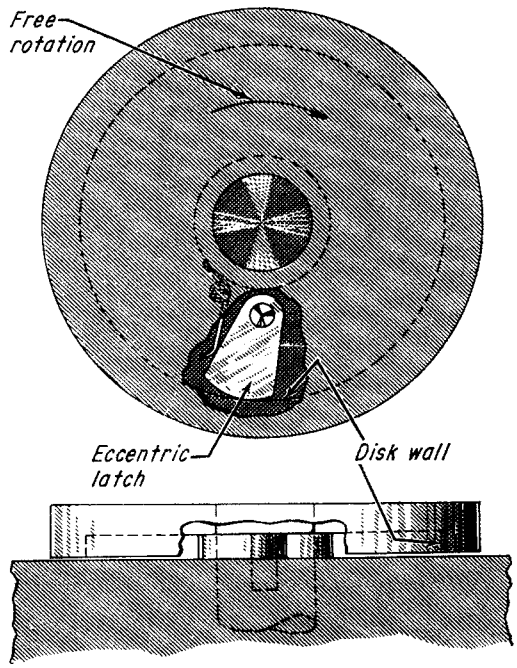

3 **SINGLE-TOOTH RATCHET** has special pawl, shaped to avoid annoying sound. The notch corner engages pawl before it can hit the flat.



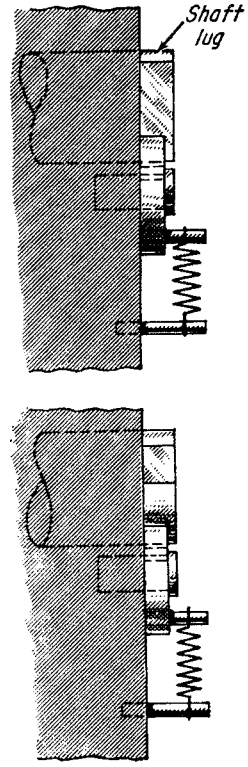
4 **SWINGING PAWL** will operate silently in any position. As one pawl tooth leaves the ratchet, the other tooth engages at about half depth.



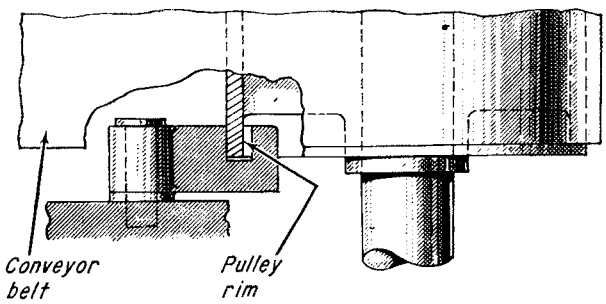

6 **SLIDING PIN** is moved into position (during normal rotation) to prevent reverse rotation. Key the cam stop into housing.

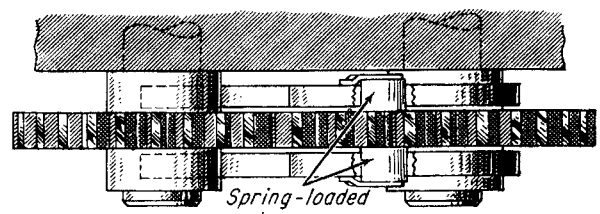


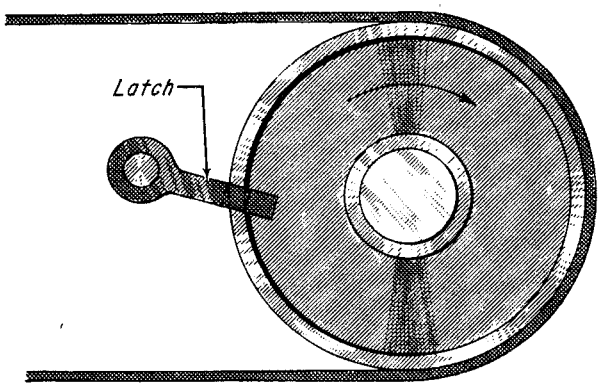

7 ECCENTRIC LATCH allows shaft to rotate in one direction; attempted reversal immediately causes latch to wedge against disk wall.

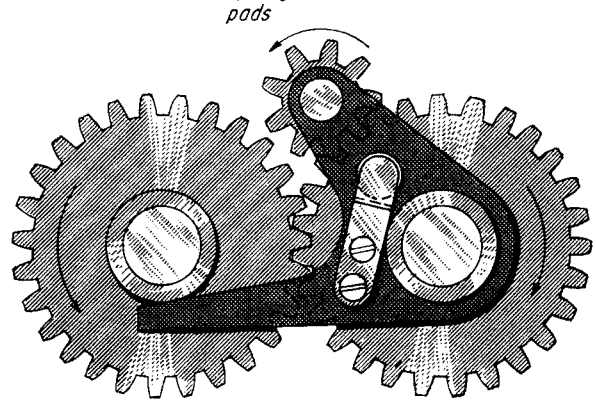



8



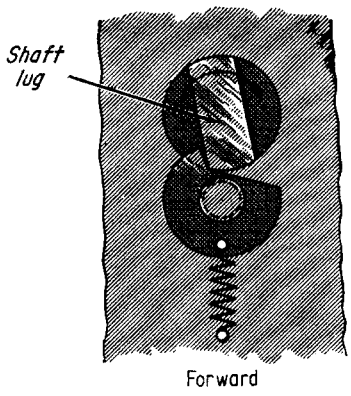



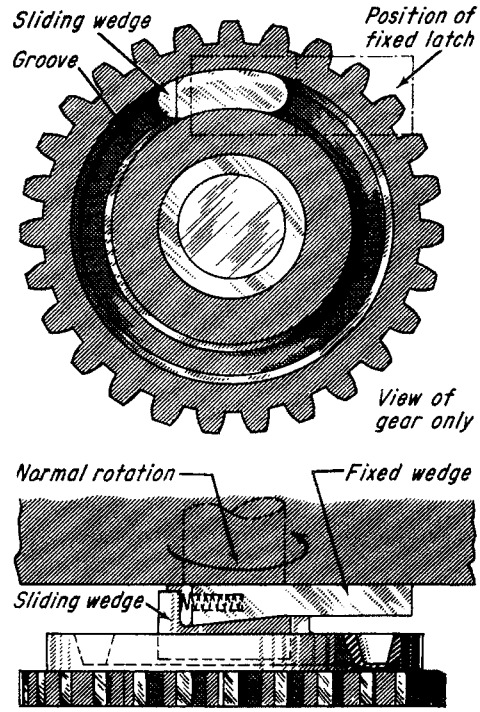



10 LATCH ON RIM of pulley is free only when rotation is in direction shown. This arrangement is ideal for conveyor-belt pulleys.

11 SPRING-LOADED FRICTION PADS contact the right gear. Idler meshes and locks gear set when rotation is reversed.

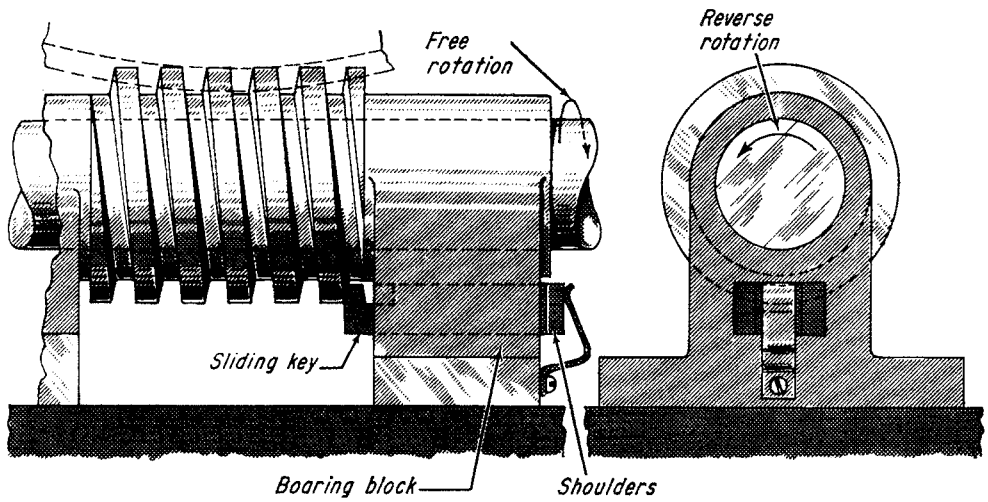



**LUG ON SHAFT** pushes the notched disk free during normal rotation. Disk periphery stops lug to prevent reverse rotation.




9

**FIXED WEDGE AND SLIDING WEDGE** tend to disengage when the gear is turning clockwise. The wedges jam in reverse direction.




12

**SLIDING KEY** has tooth which engages the worm threads. In reverse rotation key is pulled in until its shoulders contact block.

# Automatic Stopping Mechanisms for Faulty Machine Operation

**Many machines, particularly automatically operated production machines, may damage themselves or parts being processed unless they are equipped with devices that stop the machine or cause it to skip an operation when something goes wrong. The accompanying patented mechanisms show principles that can be employed to interrupt normal machine operation: Mechanical, electrical or electronic, hydraulic, pneumatic, or combinations of these means. Endless varieties of each method are in use.**

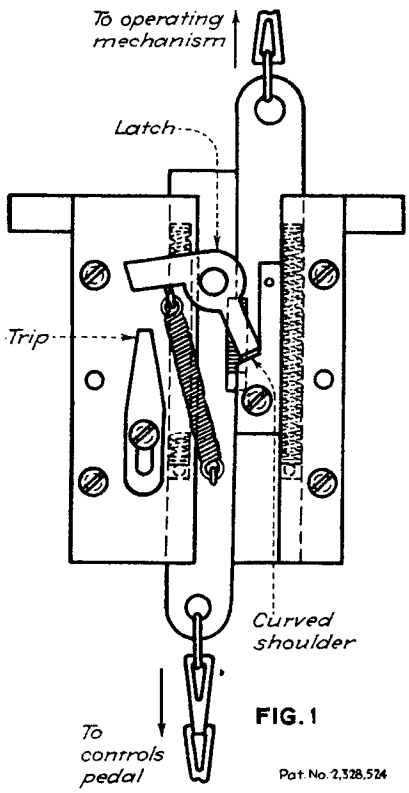



FIG. 1

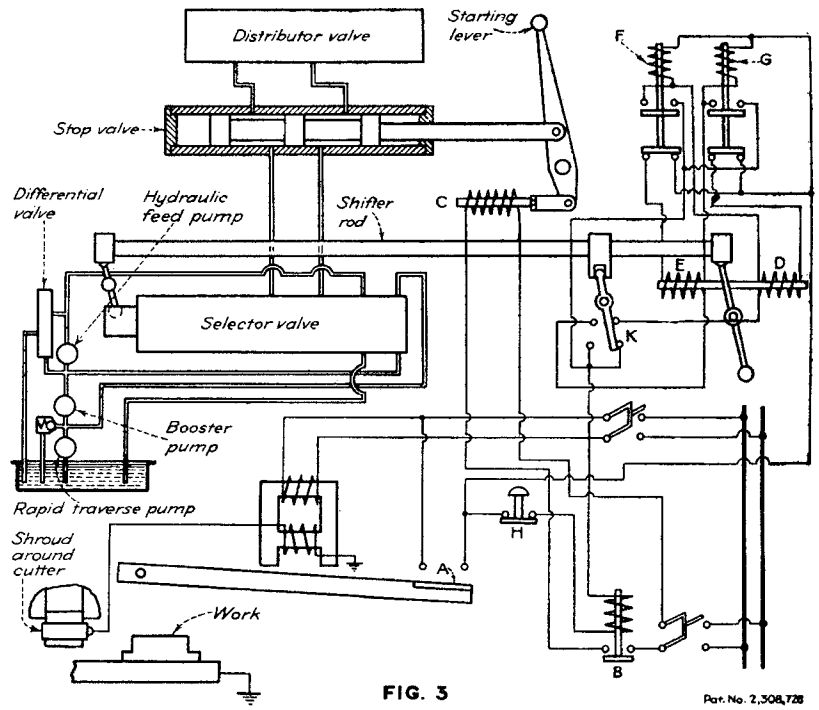



FIG. 3

**Fig. 1**—Repetition of machine cycle is prevented if pedal remains depressed. Latch carried by left slide pushes right slide downward by means of curved shoulder until latch is disengaged by trip member.

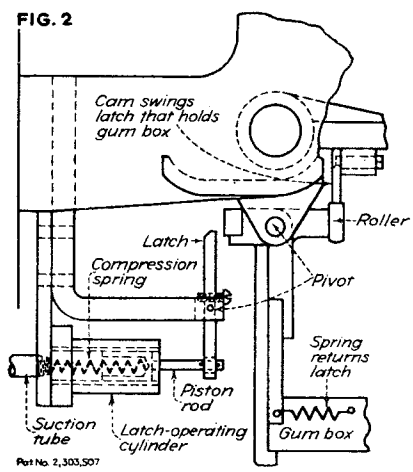



FIG. 2

**Fig. 2**—Gumming of suction picker and label carrier when label is not picked up by the suction, is prevented by insufficient suction on latch-operating cylinder, caused by open suction holes on picker. When latch-operating cylinder does not operate, gum box holding latch returns to holding position after cyclic removal by cam and roller, thus preventing gum box and rolls from rocking to make contact with picker face.

**Fig. 3**—Damage to milling cutter, work or fixtures is prevented by shroud around cutter, which upon contact closes electric circuit through relay, thus closing contact *A*. This causes contact *B* to close, thus energizing relay *C* to operate stop valve, and closes circuit through relay *D*, thus reversing selector valve by means of shifter rod so that bed travel will reverse on starting. Simultaneously, relay *F* opens circuit of relay *E* and closes a holding circuit that was broken by the shifter lever at *K*. Relay *G* also closes a holding circuit and opens circuit through relay *D*. Starting lever, released by push button *H*, releases contact *A* and returns circuit to normal. If contact is made with shroud when bed travel is reversed, interchange *D* and *E*, and *F* and *G* in above sequence of operations.

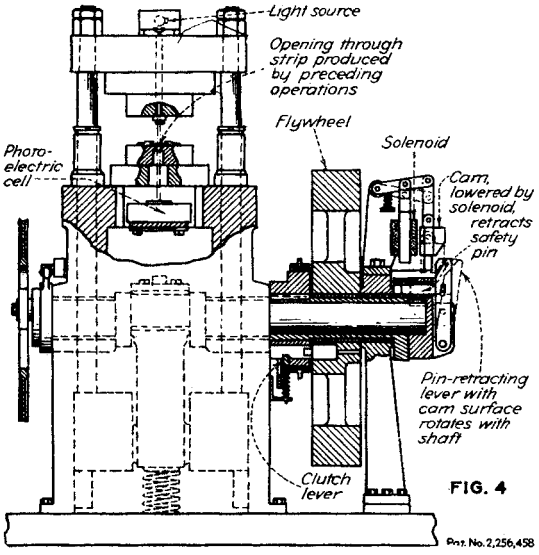



FIG. 4

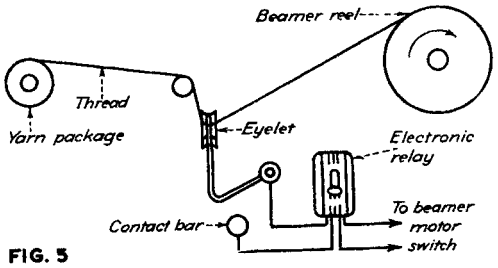



FIG. 5

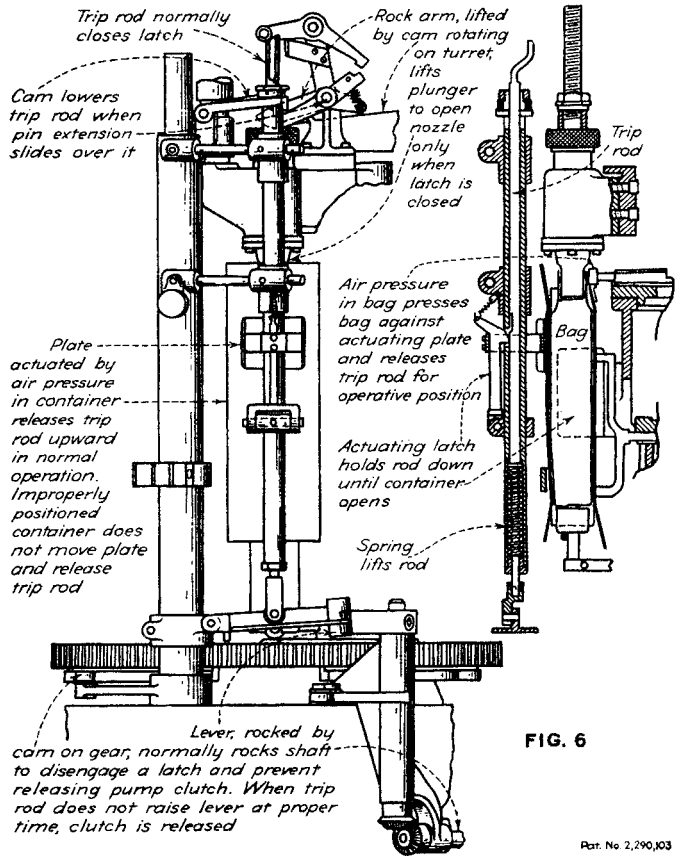

Trip rod normally closes latch
Rock arm, lifted by cam rotating on turret, lifts plunger to open nozzle only when latch is closed
Cam lowers trip rod when pin extension slides over it
Trip rod
Air pressure in bag presses bag against actuating plate and releases trip rod for operative position
Bag
Plate actuated by air pressure in container releases trip rod upward in normal operation. Improperly positioned container does not move plate and release trip rod
Actuating latch holds rod down until container opens
Spring lifts rod
Lever, rocked by cam on gear, normally rocks shaft to disengage a latch and prevent releasing pump clutch. When trip rod does not raise lever at proper time, clutch is released
Pat. No. 2,290,103

FIG. 6

**Fig. 4**—High-speed press is stopped when metal strip advances improperly so that hole punched in strip fails to match with opening in die block to permit passage of light beam. Intercepted light beam to photo-electric cell results in energizing solenoid and withdrawal of clutch pin.

**Fig. 5**—Broken thread permits contact bar to drop, thereby closing electronic relay circuit, which operates to stop beamer reeling equipment.

**Fig. 6**—Nozzle on packaging machine does not open when container is not in proper position.

**Fig. 7**—Obstruction under explorer foot of wire-stitching machine prevents damage to machine by raising a vertical plunger, which releases a latch lever so that rotary cam raises lever that retains clutch operating plunger.

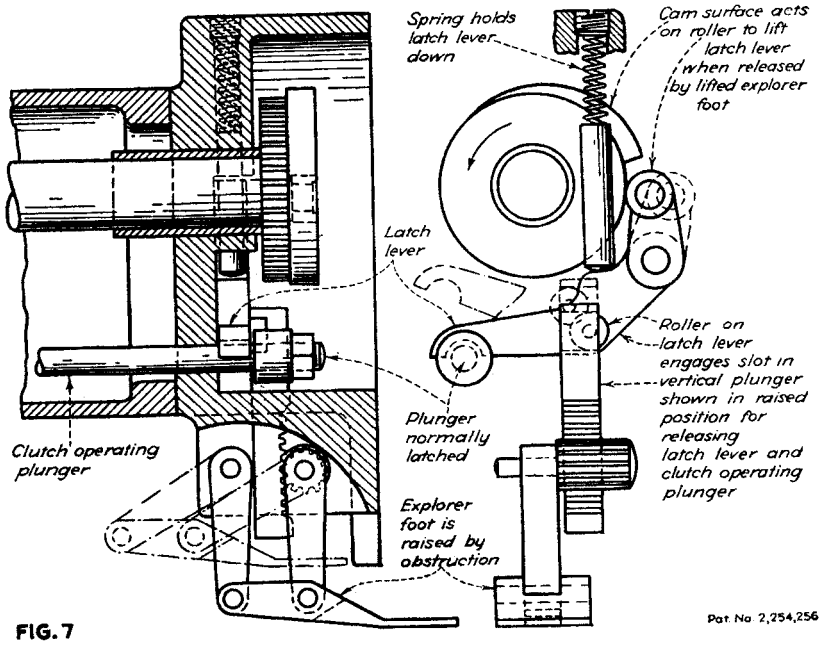

Spring holds latch lever down
Cam surface acts on roller to lift latch lever when released by lifted explorer foot
Latch lever
Roller on latch lever engages slot in vertical plunger shown in raised position for releasing latch lever and clutch operating plunger
Plunger normally latched
Clutch operating plunger
Explorer foot is raised by obstruction
Pat. No. 2,254,256

FIG. 7

# Automatic Stop Mechanisms Protect Machines and Work

**This group of stop mechanisms includes types and applications for machines in different fields. Such devices, which prevent automatic machines form damaging themselves or the work passing through them, make use of mechanical, electrical, hydraulic, and pneumatic principles. Typical mechanisms illustrated prevent excess speed, miswearing, jamming of toggle press and foodcanning machines, operation of printing presses when the paper web breaks, improper feeding of wrapping paper, and uncoordinated operation of glass-making machines.**

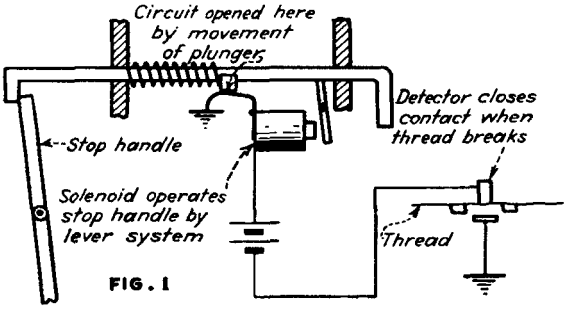

Circuit opened here by movement of plunger
Detector closes contact when thread breaks
Stop handle
Solenoid operates stop handle by lever system
Thread

FIG. 1

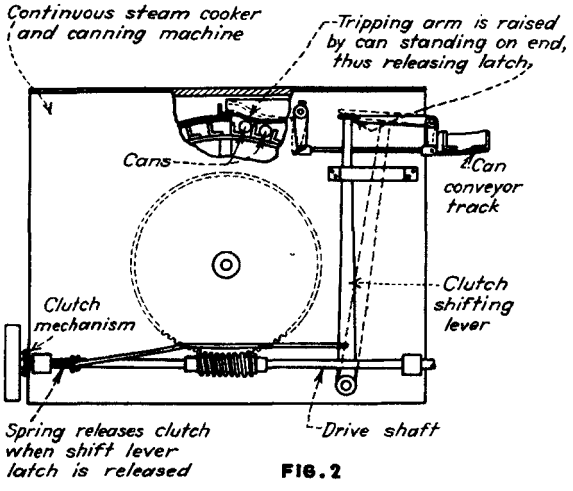

Continuous steam cooker and canning machine
Tripping arm is raised by can standing on end, thus releasing latch
Cans
Can conveyor track
Clutch shifting lever
Clutch mechanism
Spring releases clutch when shift lever latch is released
Drive shaft

FIG. 2

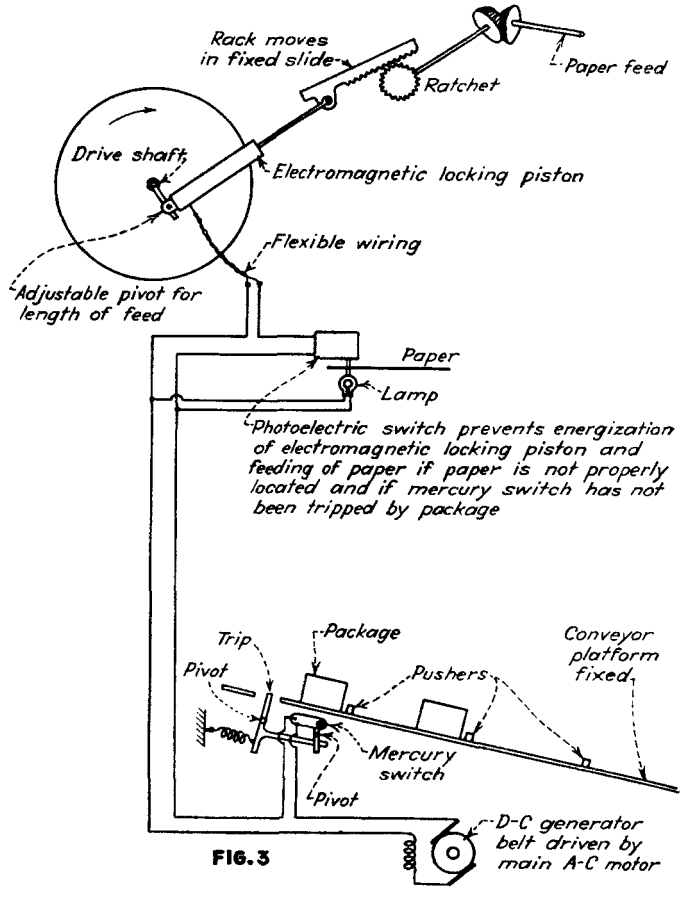



FIG. 3

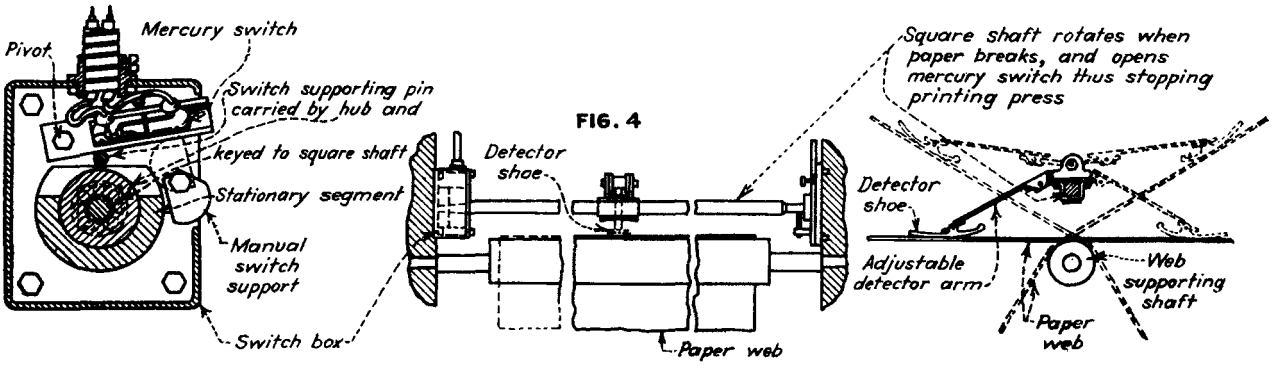



FIG. 4

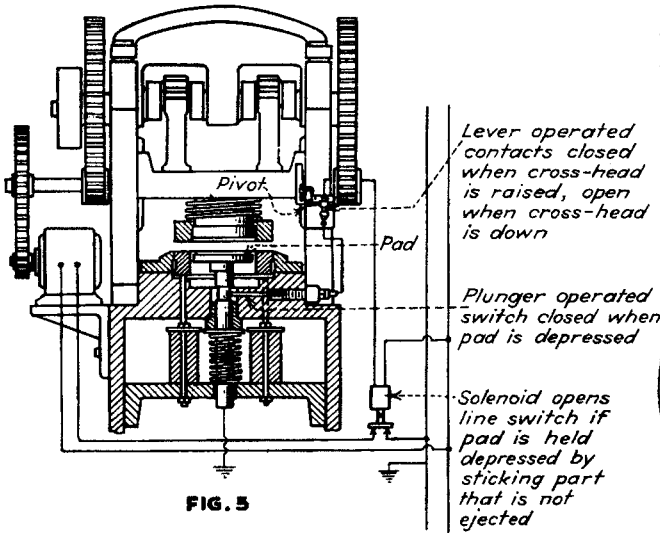

Pivot
Pad
Lever operated contacts closed when cross-head is raised, open when cross-head is down
Plunger operated switch closed when pad is depressed
Solenoid opens line switch if pad is held depressed by sticking part that is not ejected

FIG. 5

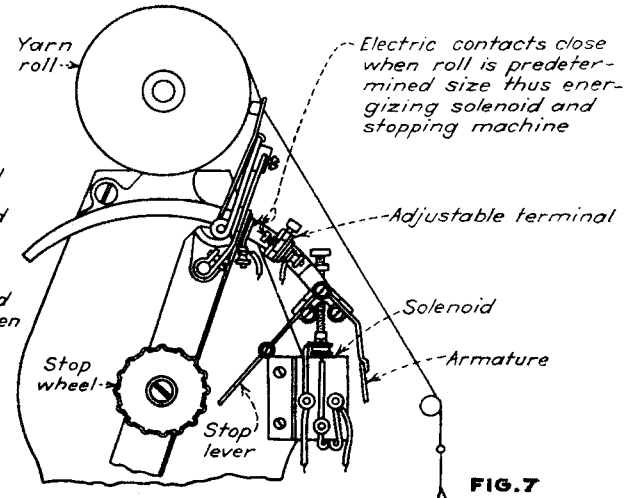



FIG. 7

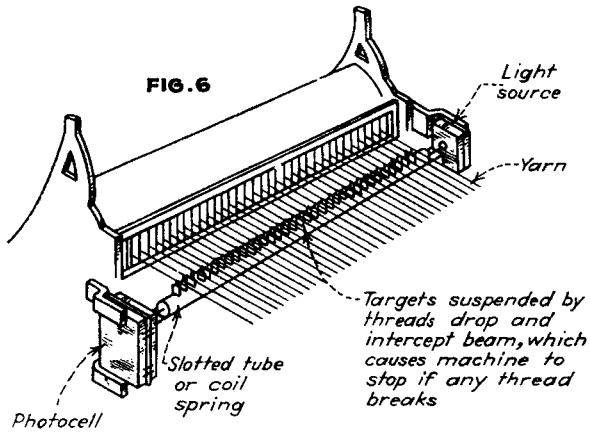
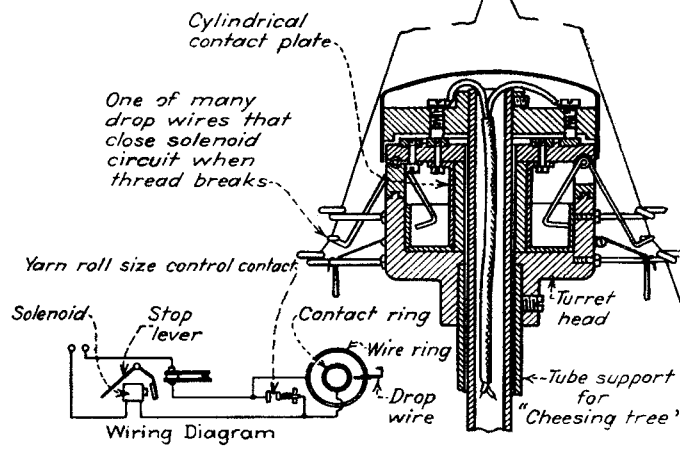

Light source
Yarn
Targets suspended by threads drop and intercept beam, which causes machine to stop if any thread breaks
Slotted tube or coil spring
Photocell

FIG. 6



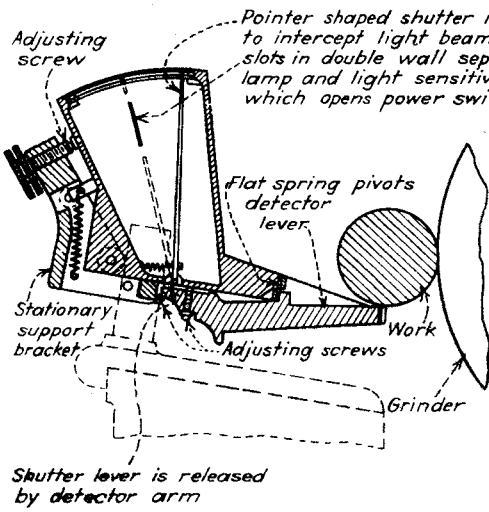


Adjusting screw
Pointer shaped shutter moves to intercept light beams through slots in double wall separating lamp and light sensitive cell, which opens power switch
Flat spring pivots detector lever
Stationary support bracket
Work
Adjusting screws
Grinder
Shutter lever is released by detector arm

FIG. 8

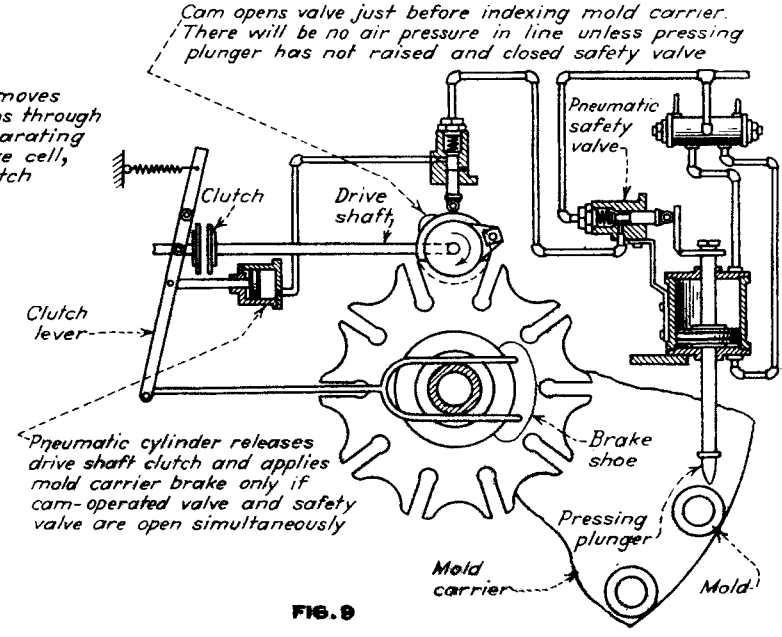

Cam opens valve just before indexing mold carrier. There will be no air pressure in line unless pressing plunger has not raised and closed safety valve
Pneumatic safety valve
Clutch
Drive shaft
Clutch lever
Pneumatic cylinder releases drive shaft clutch and applies mold carrier brake only if cam-operated valve and safety valve are open simultaneously
Brake shoe
Pressing plunger
Mold carrier
Mold

FIG. 9

# Mechanical Automatic Stop Mechanisms

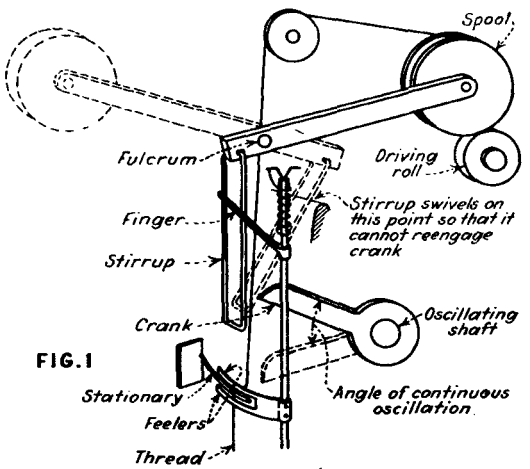



FIG. 1

FIG. 1—A mechanism used on the Barber-Colman spooler. When thread breaks the feelers are released and the spiral spring causes the spindle with finger to rotate. Latter throws the stirrup into the path of the oscillating crank, which on its downward stroke throws the spool into the position shown dotted, the stirrup then being thrown out of the path of the oscillating crank.

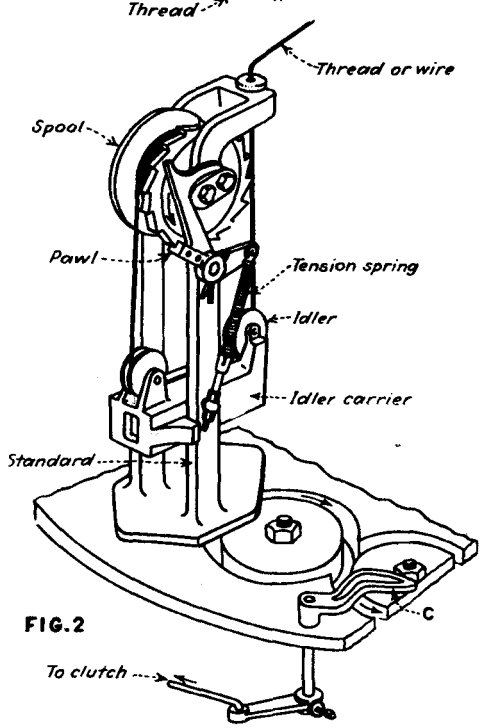



FIG. 2

FIG. 2—Mechanism used with variations on tubular braiding machines. When braiding, tension on the wire or thread lifts the idler carrier which thereby releases the pawl from the ratchet on spool flange and allows the spool to turn and unwind. When machine stops the tension on wire is decreased allowing the idler carrier to fall so that the pawl can engage the ratchet. If wire breaks while the machine is running the unsupported idler carrier falls to the base of the standard and when the standard arrives at the station in the raceway adjacent to the cam *C*, the lug *L* on idler carrier strikes the cam *C*, rotating it far enough to disengage a clutch on the driving shaft, thereby stopping the machine.

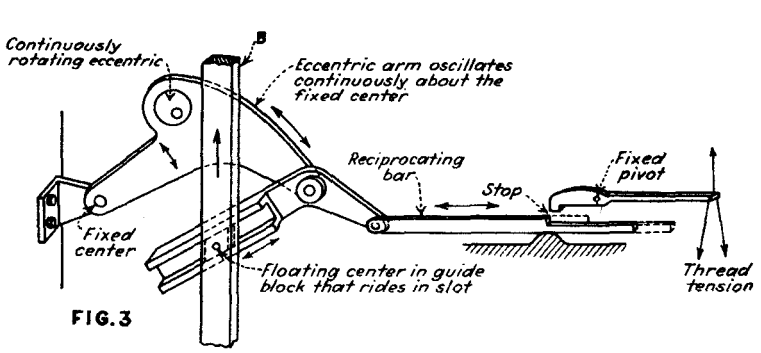



FIG. 3

FIG. 3—When thread breaks the stop drops and intercepts reciprocating bar. On the next counter-clockwise oscillation of the eccentric arm the bar *B* is raised. A feature of this design is that it permits the arm *B* to move up or down independently for a limited distance.

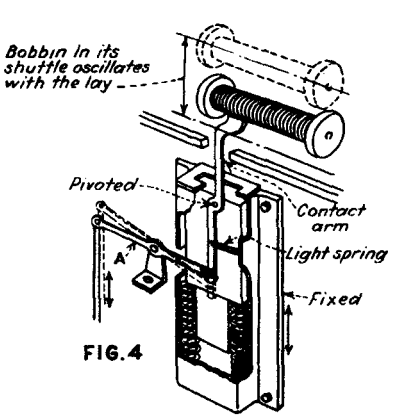



FIG. 4

FIG. 4—Schematic diagram of mechanism to cause bobbin changer to operate. If contact arm does not slip on bobbin the lever *A* will rotate to the position shown. But if contact with bobbin center slips, as it will do if the bobbin is empty.

**Designs shown here diagrammatically were taken from textile machines, braiding machines and packaging machines. Possible modifications of them to suit other applications will be apparent.**

lever *A* will not rotate to position indicated by dashed line, thereby causing bobbin changer to come into action.

FIG. 5—Simple type of stop mechanism for limiting the stroke of a reciprocating machine member. Arrows indicate the direction of movements.

FIG. 6—When the predetermined weight of material has been poured on the pan, the movement of the scale beam pushes the latch out of engagement, allowing the paddle wheel to rotate and thus dump the load. The scale beam drops, thereby returning the latch to the holding position and stopping the wheel when the next vane hits the latch.

FIG. 7—In this textile machine any movement that will rotate the stop lever counter-clockwise will bring it in the path of the continuously reciprocating shaft. This will cause the catch lever to be pushed counter-clockwise and the hardened steel stop on the clutch control shaft will be freed. A spiral spring then impels the clutch control shaft to rotate clockwise, which movement throws out the clutch and applies the brake. Initial movement of the stop lever may be caused by the breaking of a thread, a moving dog, or any other means.

FIG. 8—Arrangement used on some package loading machines to stop machine if a package should pass loading station without receiving an insert. Pawl finger *F* has a rocking motion obtained from crankshaft, timed so that it enters the unsealed packages and is stopped against the contents. If the box is not filled the finger enters a considerable distance and the pawl end at bottom engages and holds a ratchet wheel on driving clutch which disengages the machine driving shaft.

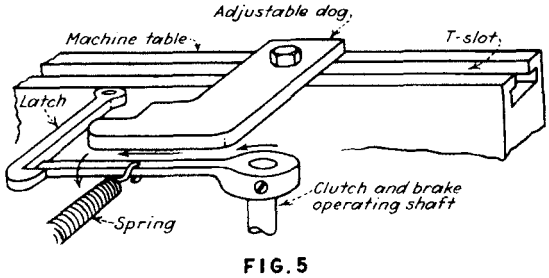



FIG. 5

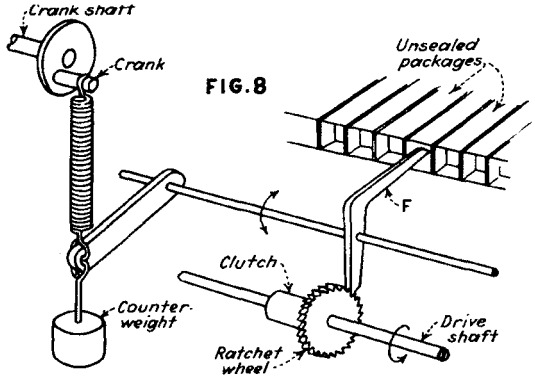



FIG. 8

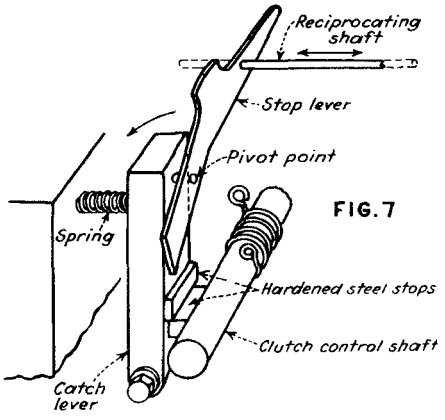



FIG. 7

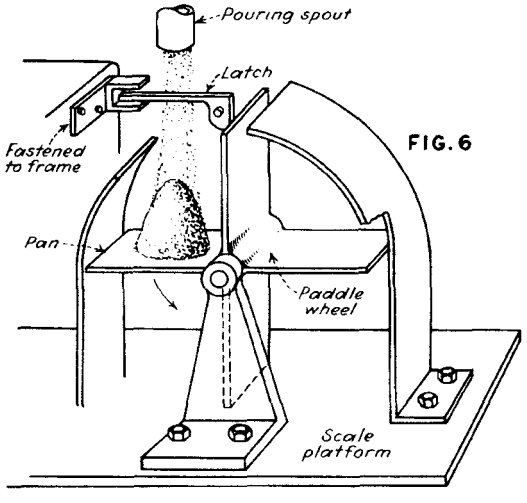



FIG. 6

# Automatic Feed Mechanisms for Various Materials

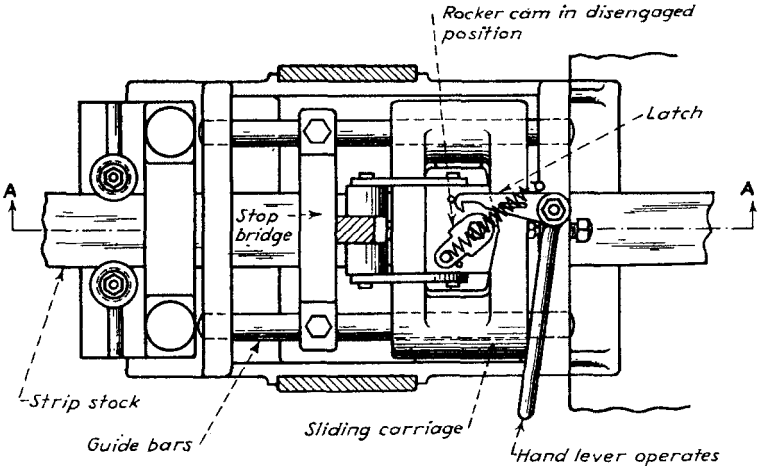
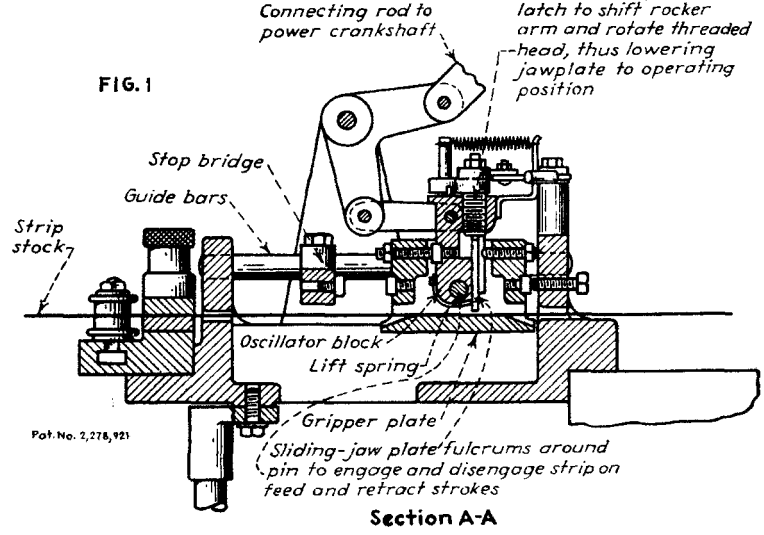
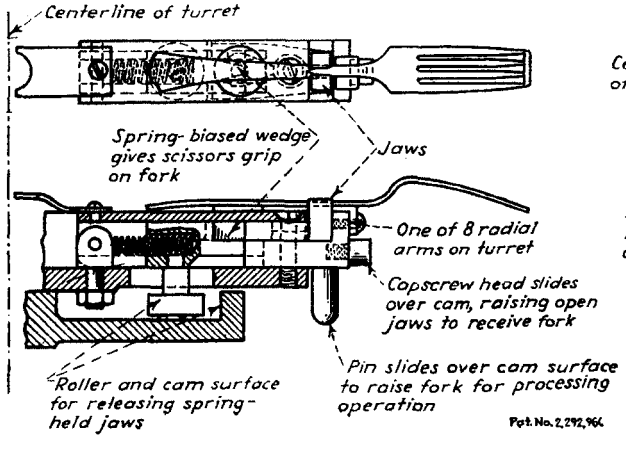



FIG. 1




Section A-A

**Design of feed mechanisms for automatic or semi-automatic machines depends largely upon such factors as size, shape, and character of materials or parts being fed into a machine, and upon the type of operations to be performed. Feed mechanisms may be merely conveyors, may give positive guidance in many instances, or may include tight holding devices if the parts are subjected to processing operations while being fed through a machine. One of the functions of feed mechanism is to extract single pieces from a stack or unassorted supply of stock or, if the stock is a continuous strip of steel, roll of paper, long bar, and the like, to maintain intermittent motion between processing operations. All of these conditions are illustrated in the accompanying feed mechanisms.**


Centerline of turret
Spring-biased wedge gives scissors grip on fork
Jaws
One of 8 radial arms on turret
Capscrew head slides over cam, raising open jaws to receive fork
Roller and cam surface for releasing spring-held jaws
Pin slides over cam surface to raise fork for processing operation
Pat. No. 2,292,964

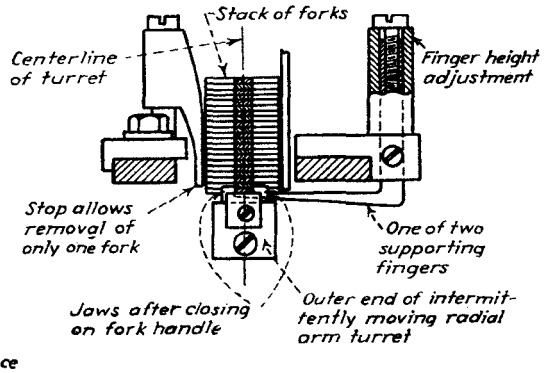

Stack of forks
Centerline of turret
Finger height adjustment
Stop allows removal of only one fork
One of two supporting fingers
Jaws after closing on fork handle
Outer end of intermittently moving radial arm turret

FIG. 2

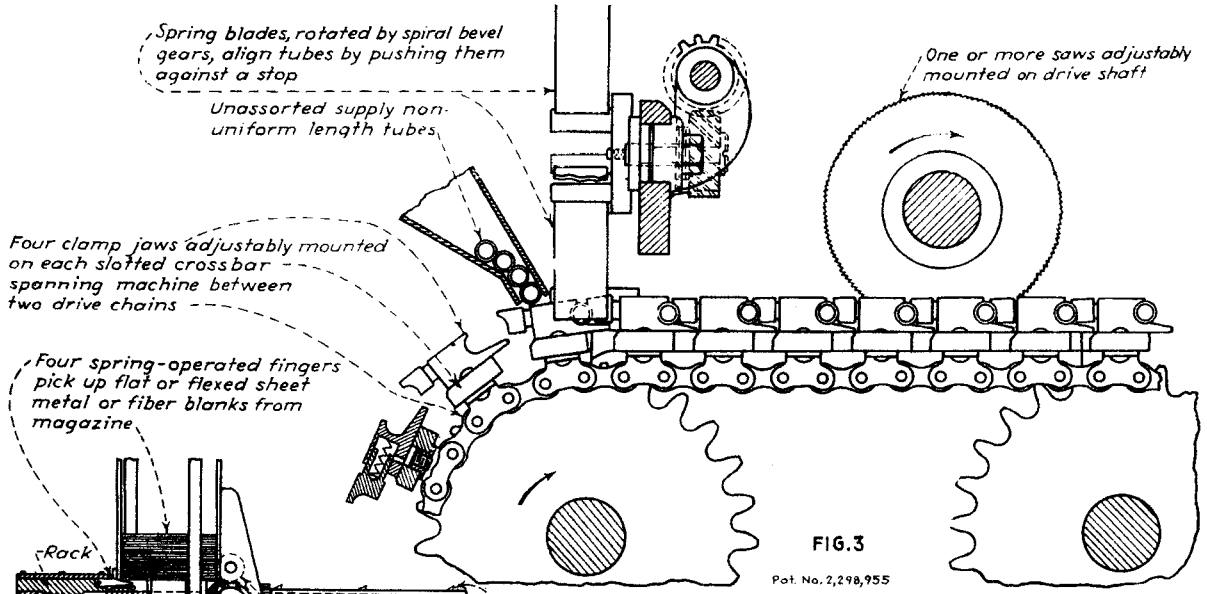

Spring blades, rotated by spiral bevel gears, align tubes by pushing them against a stop
Unassorted supply non-uniform length tubes
One or more saws adjustably mounted on drive shaft
Four clamp jaws adjustably mounted on each slotted crossbar spanning machine between two drive chains
Four spring-operated fingers pick up flat or flexed sheet metal or fiber blanks from magazine

FIG.3

Pat. No. 2,298,955

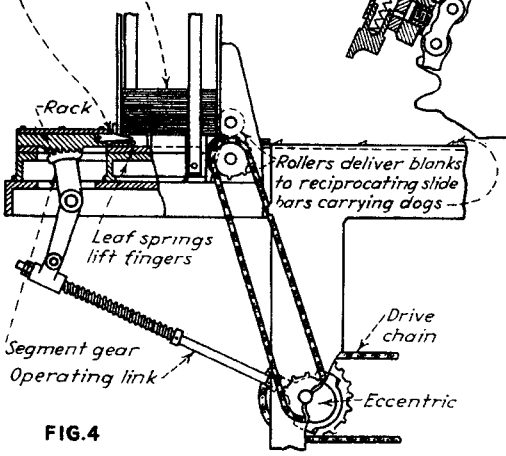
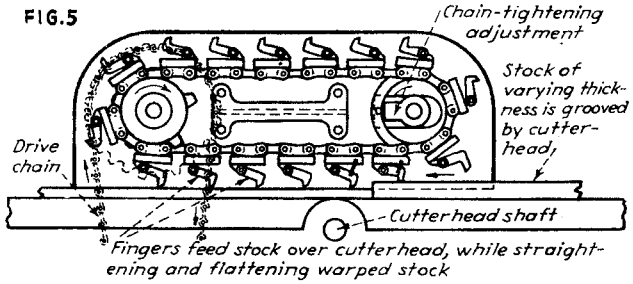



FIG.4

Pat. No. 2,270,871

FIG.5


Chain-tightening adjustment
Stock of varying thickness is grooved by cutter-head
Drive chain
Cutterhead shaft
Fingers feed stock over cutterhead, while straightening and flattening warped stock

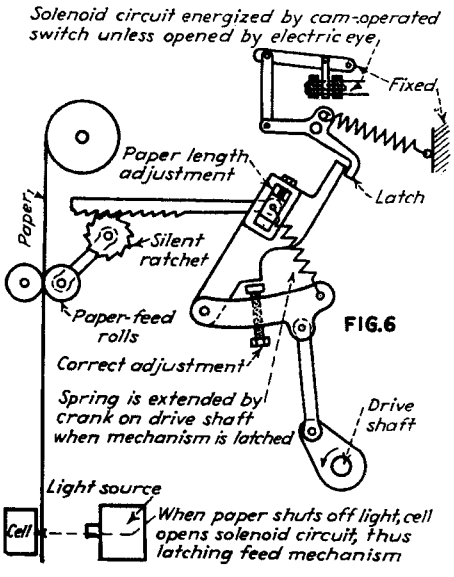

Solenoid circuit energized by cam-operated switch unless opened by electric eye
Fixed
Paper length adjustment
Latch
Paper
Silent ratchet
Paper-feed rolls
Correct adjustment
Spring is extended by crank on drive shaft when mechanism is latched
Drive shaft
Light source
Cell
When paper shuts off light, cell opens solenoid circuit, thus latching feed mechanism

FIG.6

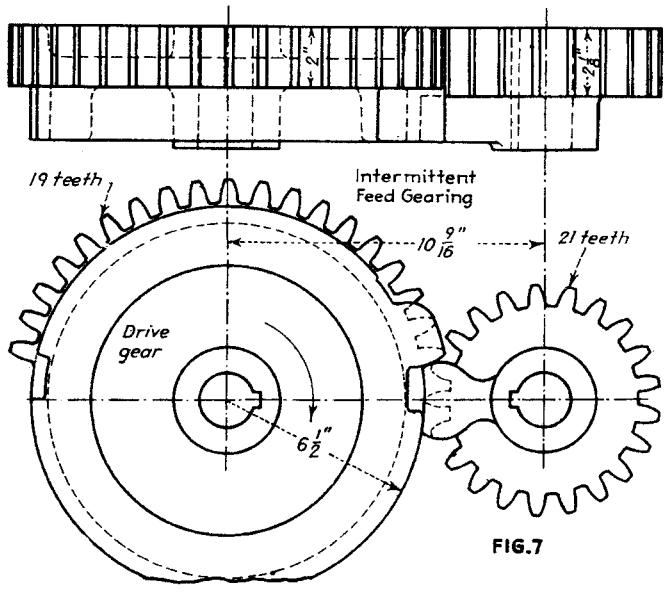



FIG.7

# Automatic Safety Mechanisms for Operating Machines

THE most satisfactory automatic guard mechanisms for preventing injury to machine operators are those that have been designed with the machine. When properly designed they (1) do not reduce visibility, (2) do not impede the operator, (3) do not cause painful blows on the operator's hand in avoiding serious injury, (4) are safe with respect to wear in the safety mechanism, (5) are sensitive and instantaneous in operation, and (6) render the machine inoperative if tampered with or removed.

Safety devices range from those that keep both hands occupied with controls away from the work area to guards that completely inclose the work during operation of the machine and prevent operation of the machine unless so protected. The latter might include the "electric eye," which is the activating means of one of the mechanisms illustrated.

FIG.1

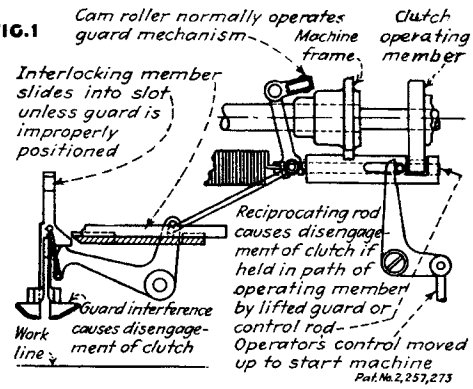
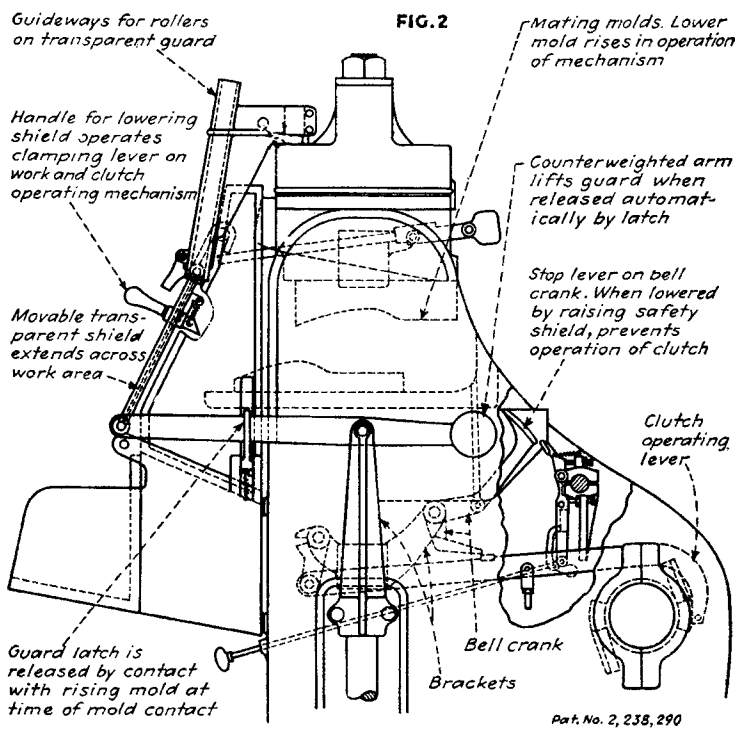

Cam roller normally operates guard mechanism
Machine frame
Clutch operating member
Interlocking member slides into slot unless guard is improperly positioned
Reciprocating rod causes disengagement of clutch if held in path of operating member by lifted guard or control rod
Guard interference causes disengagement of clutch
Work line
Operators control moved up to start machine
Pat.No.2,257,273

FIG.2


Guideways for rollers on transparent guard
Handle for lowering shield operates clamping lever on work and clutch operating mechanism
Movable transparent shield extends across work area
Guard latch is released by contact with rising mold at time of mold contact
Mating molds. Lower mold rises in operation of mechanism
Counterweighted arm lifts guard when released automatically by latch
Stop lever on bell crank. When lowered by raising safety shield, prevents operation of clutch
Clutch operating lever
Bell crank
Brackets
Pat. No. 2, 238, 290

Perspective of Slide

Pat. No. 2,301,817

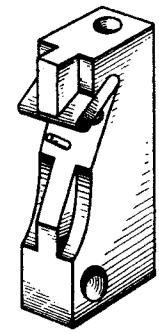
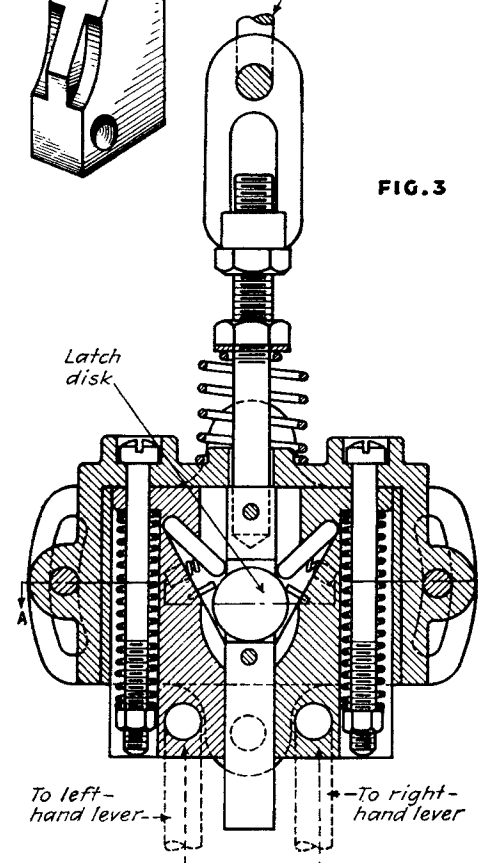
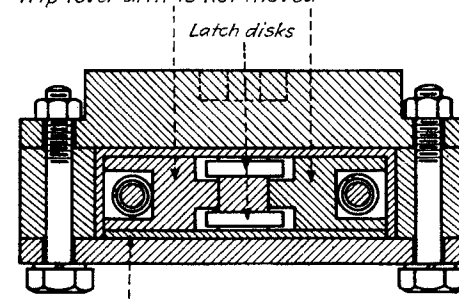


FIG.3




Unless both slide blocks are moved and engage two latch disks simultaneously, disks are displaced into races of slide block and trip lever arm is not moved




Section A-A

Box assembly slides in stationary housing when slide blocks move together

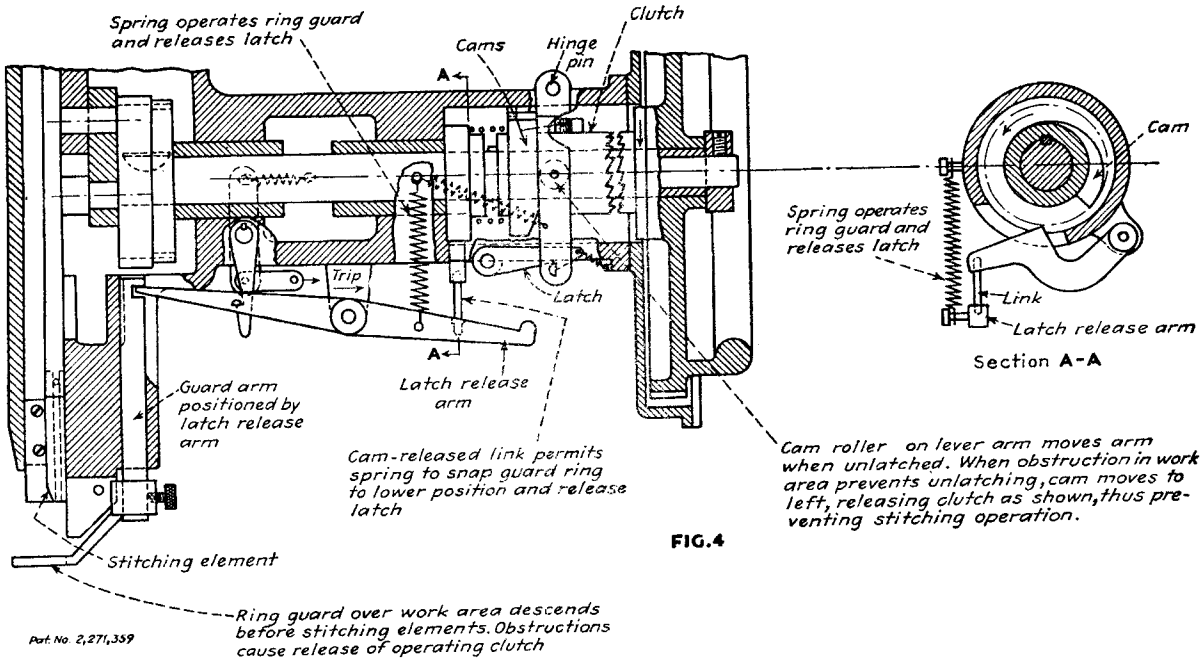

Spring operates ring guard and releases latch
Cams
Hinge pin
Clutch
A
Trip
Latch
A
Guard arm positioned by latch release arm
Latch release arm
Cam-released link permits spring to snap guard ring to lower position and release latch
Stitching element
Ring guard over work area descends before stitching elements. Obstructions cause release of operating clutch
Pat. No. 2,271,359
Cam
Spring operates ring guard and releases latch
Link
Latch release arm
Section A-A
Cam roller on lever arm moves arm when unlatched. When obstruction in work area prevents unlatching, cam moves to left, releasing clutch as shown, thus preventing stitching operation.

FIG.4

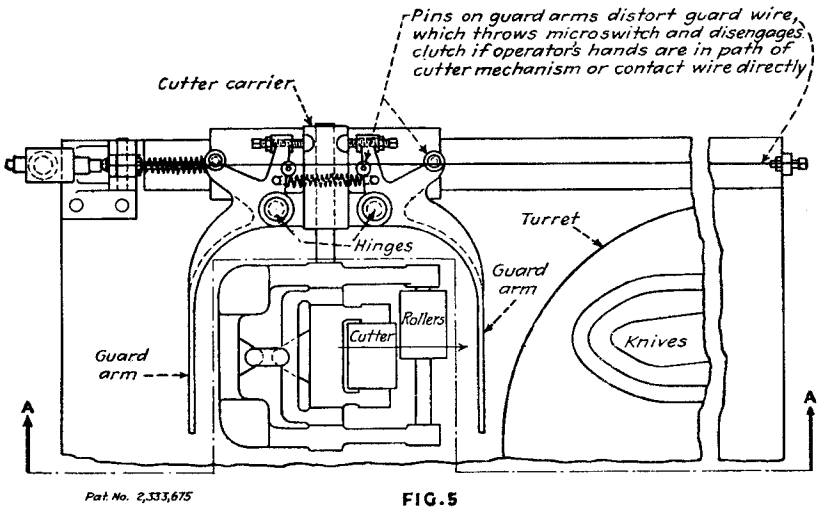
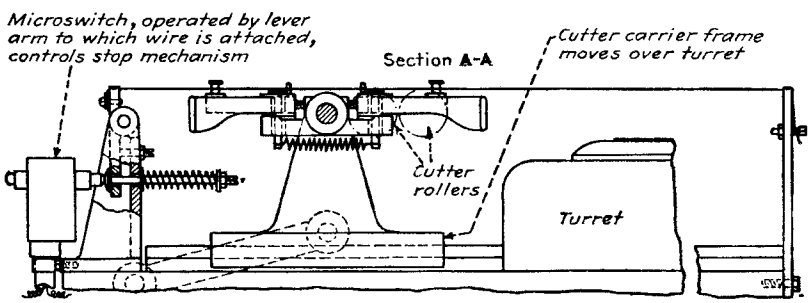



FIG.5


Microswitch, operated by lever arm to which wire is attached, controls stop mechanism
Section A-A
Cutter carrier frame moves over turret
Cutter rollers
Turret

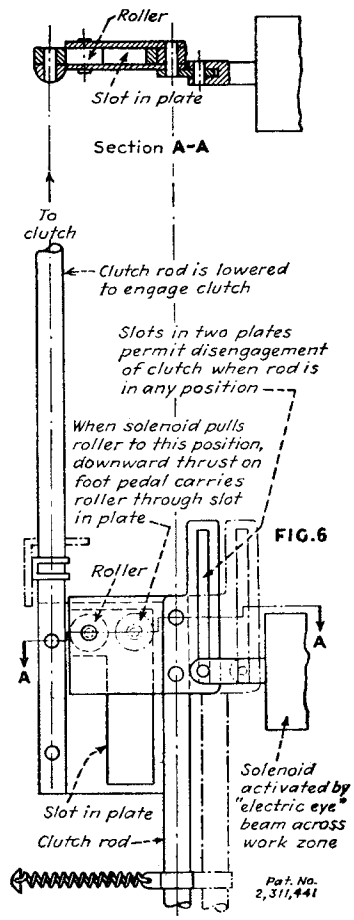

Roller
Slot in plate
Section A-A
To clutch
Clutch rod is lowered to engage clutch
Slots in two plates permit disengagement of clutch when rod is in any position
When solenoid pulls roller to this position, downward thrust on foot pedal carries roller through slot in plate
Roller
A
A
Solenoid activated by "electric eye" beam across work zone
Slot in plate
Clutch rod
Pat. No. 2,311,441

FIG.6

# Mechanisms Actuated by Air or Hydraulic Cylinders

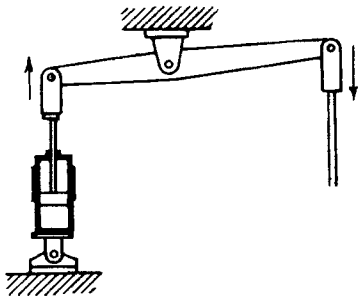


Fig. 1—Cylinder can be used with a first class lever.

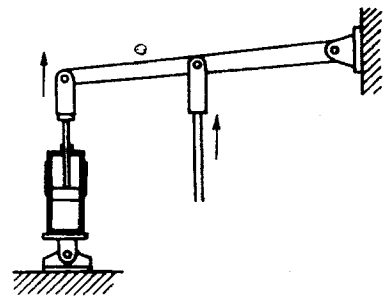


Fig. 2—Cylinder can be used with a second class lever.

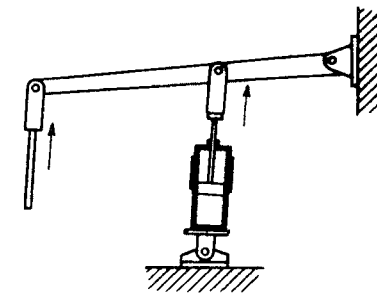


Fig. 3—Cylinder can be used with a third class lever.

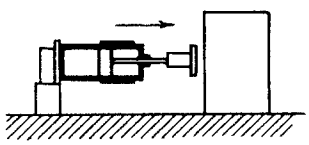


Fig. 4—Cylinder can be linked up directly to the load.

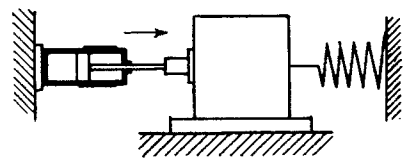


Fig. 5—Spring reduces the thrust at the end of the stroke.

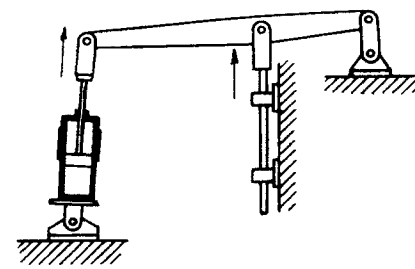


Fig. 6—Point of application of force follows the direction of thrust.

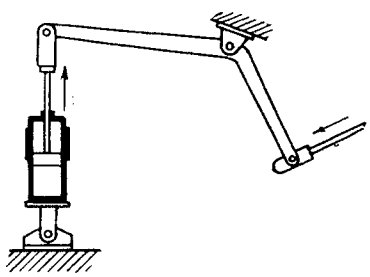


Fig. 7—Cylinder can be used with a bent lever.

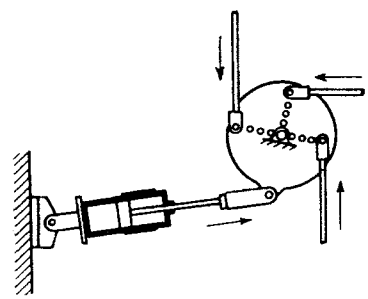


Fig. 8—Cylinder can be used with a trammel plate.

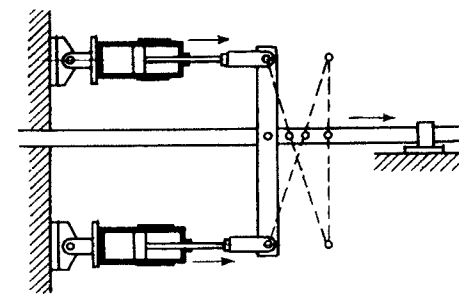


Fig. 9—Two pistons with fixed strokes position load in any of four stations.

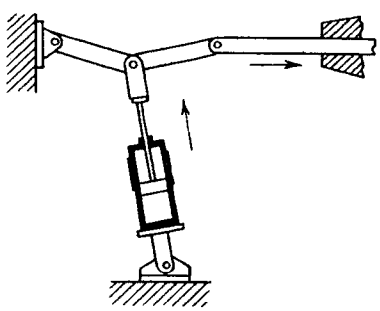


Fig. 10—A toggle can be actuated by the cylinder.

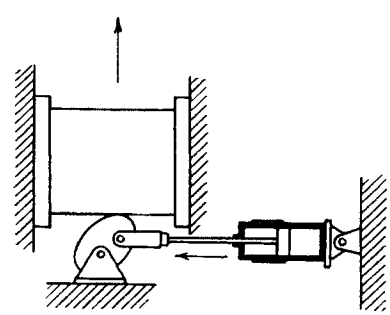


Fig. 11—The cam supports the load after completion of the stroke.

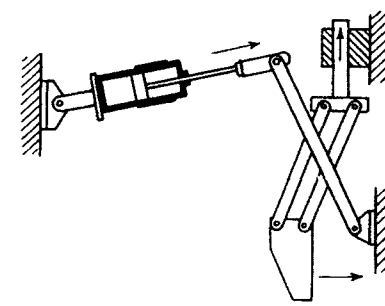


Fig. 12—Simultaneous thrusts in two different directions are obtained.

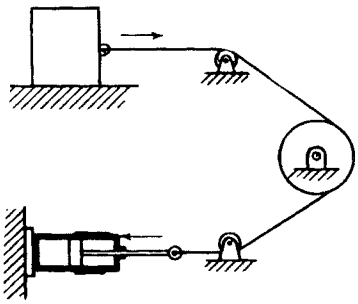


Fig. 13—Force is transmitted by a cable.

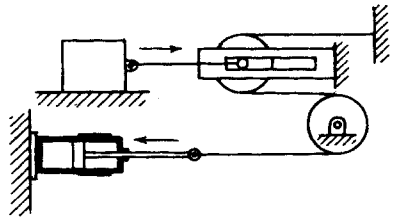


Fig. 14—Force can be modified by a system of pulleys.

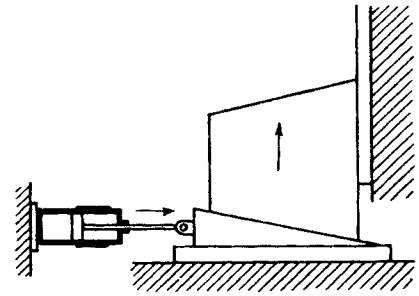


Fig. 15—Force can be modified by wedges.

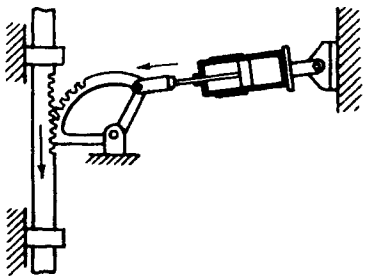


Fig. 16—Gear sector moves rack perpendicular to stroke of piston.

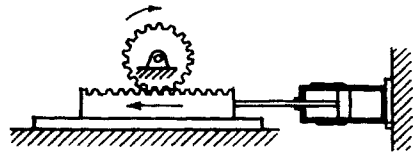


Fig. 17—Rack turns gear sector.

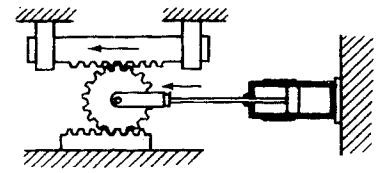


Fig. 18—Motion of movable rack is twice that of piston.

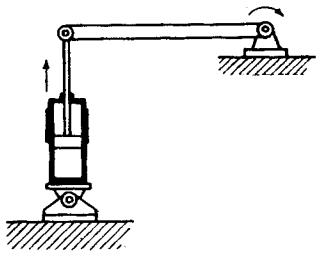


Fig. 19—Torque applied to the shaft can be transmitted to a distant point.

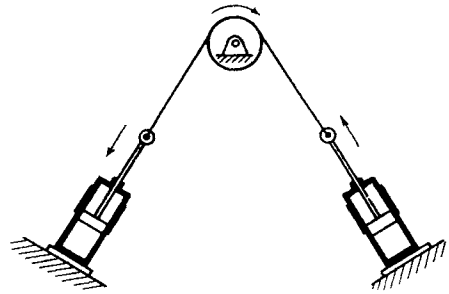


Fig. 20—Torque can also be applied to a shaft by a belt and pulley.

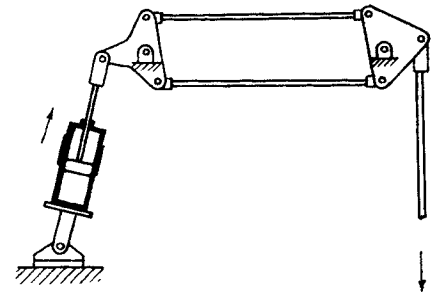


Fig. 21—Motion is transmitted to a distant point in the plane of motion.

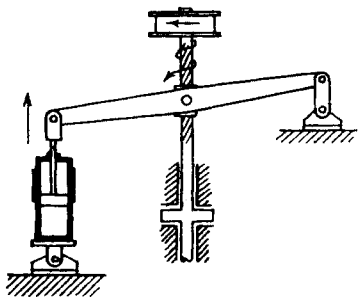


Fig. 22—A steep screw nut produces a rotation of shaft.

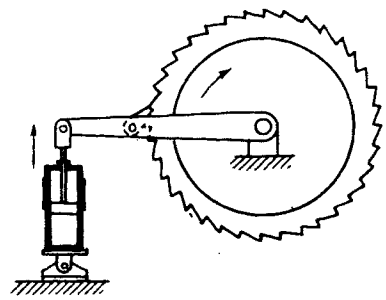


Fig. 23—Single sprocket wheel produces rotation in the plane of motion.

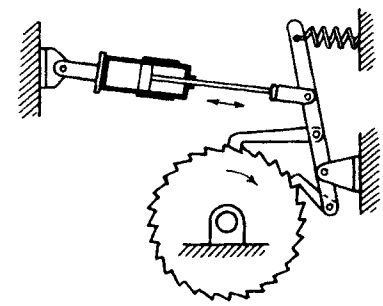
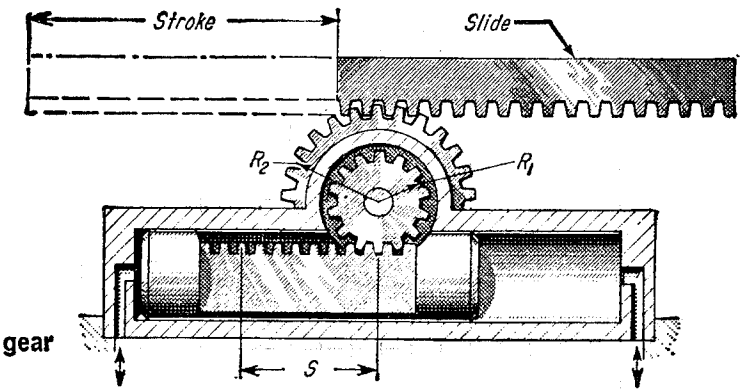


Fig. 24—Double sprocket wheel makes the rotation more nearly continuous.

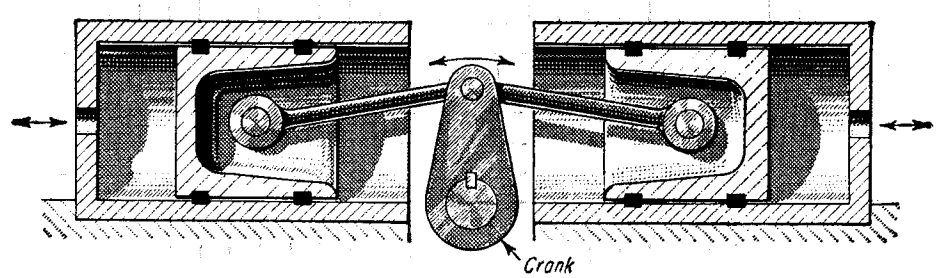
# Piston-Actuated Mechanisms

**They are also hydraulic to mechanical transducers. Output–stroke, force, speed–can be altered by a variety of methods.**

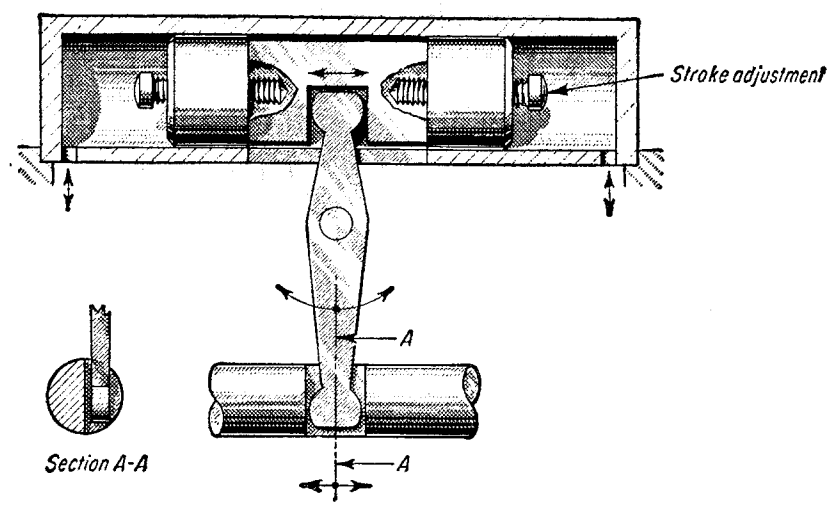
Louis Dodge



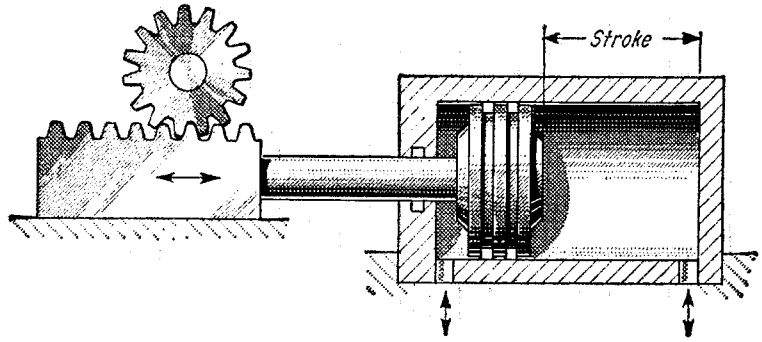

**2** **Slide stroke, force, and speed depend on gear cluster ratio**



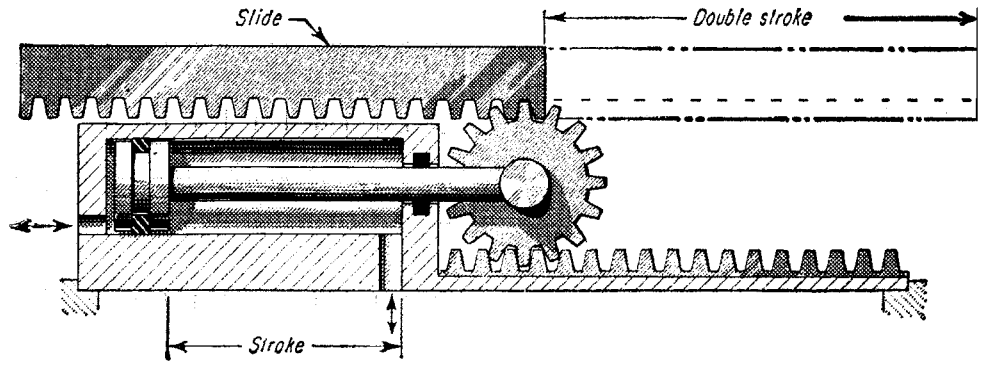

**4** **Rotary motion by alternately activated pistons**



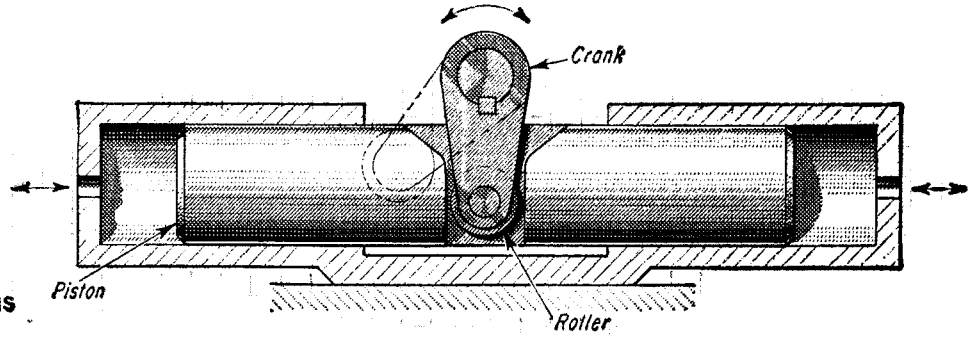

**6** **Oscillating motion with stroke adjustment**



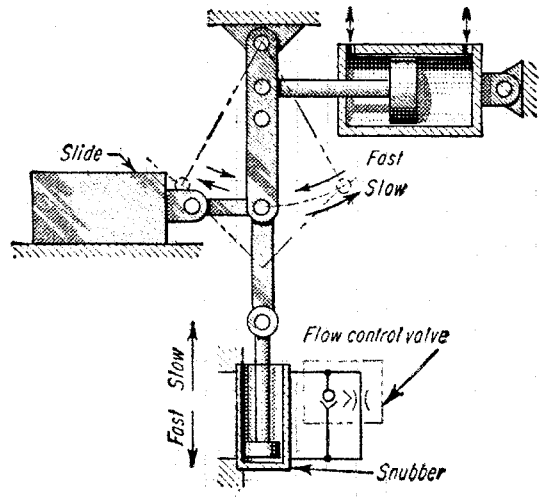

**1** Simple rack and pinion converter




**3** Gear can double speed and stroke, halve force




**5** Rotary motion without transverse load on pistons




**7** Two-speed slider from single piston

# Computing Mechanisms I

**Analog computing mechanisms are capable of virtually instantaneous response to minute variations in input. Basic units, similar to the types shown, are combined to form the final computer. These mechanisms add, subtract, resolve vectors, or solve special or trigonometric functions. Other computing mechanisms for multiplying, dividing, differentiating or integrating ore presented on the next page.**

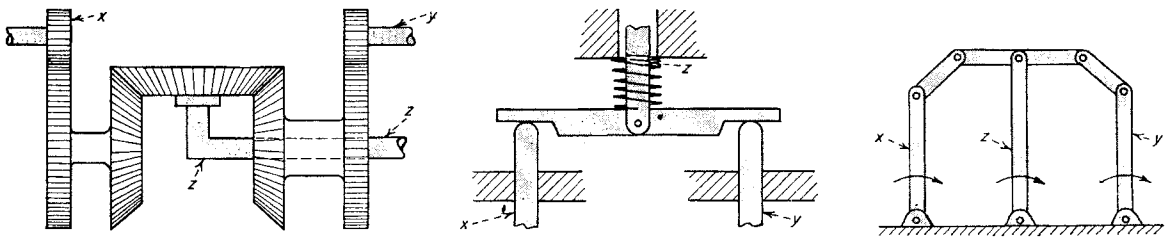



**Fig. 1—ADDITION AND SUBTRACTION.** Usually based on the differential principle; variations depend on whether inputs: (A) rotate shafts, (B) translate links, (C) angularly displaced links. Mechanisms solve equation: $z=c\ (x \pm y)$, where c is scale factor, $x$ and $y$ are inputs, and $z$ is the output. Motion of $x$ and $y$ in same direction results in addition; opposite direction—subtraction. Nineteen additional variations are illustrated in "Linkage Layouts by Mathematical Analysis," Alfred Kuhlenkamp, Product Engineering, page 165, August 1955.

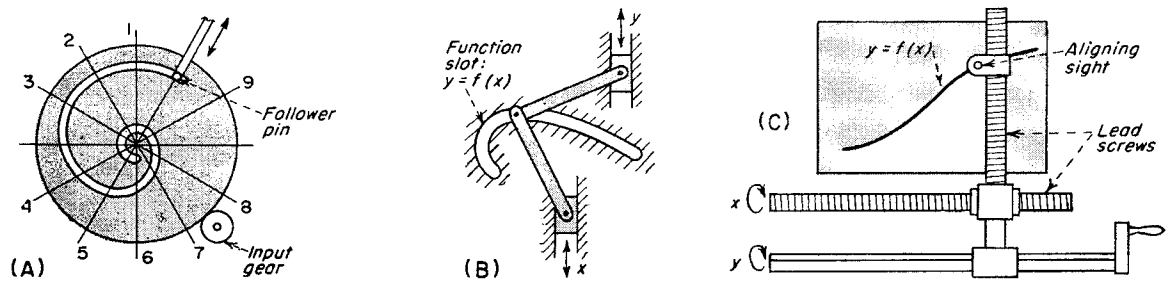



**Fig. 2—FUNCTION GENERATORS** mechanize specific equations. (A) Reciprocal cam converts a number into its reciprocal. This simplifies division by permitting simple multiplication between a numerator and its denominator. Cam rotated to position corresponding to denominator. Distance between center of cam to center of follower pin corresponds to reciprocal. (B) Function-slot cam. Ideal for complex functions involving one variable. (C) Input table. Function is plotted on large sheet attached to table. Lead screw for $x$ is turned at constant speed by an analyzer. Operator or photoelectric follower turns $y$ output to keep sight on curve.

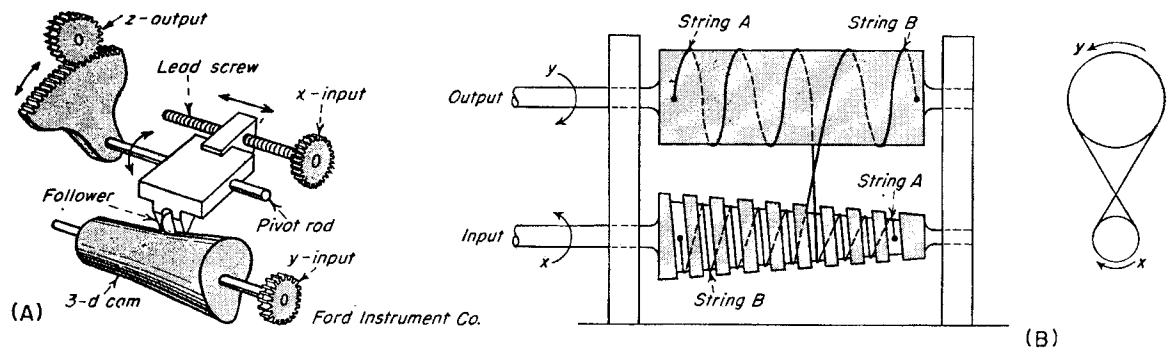



**Fig. 3—(A) THREE-DIMENSIONAL CAM** generates functions with two variables: $z=f\ (x,\ y)$. Cam rotated by $y$-input; $x$-inputs shifts follower along pivot rod. Contour of cam causes follower to rotate giving angular displacement to $z$-output gear. (B) Conical cam for squaring positive or negative inputs: $y=c\ (\pm x)^2$. Radius of cone at any point is proportional to length of string to right of point; therefore, cylinder rotation is proportional to square of cone rotation. Output is fed through a gear differential to convert to positive number.

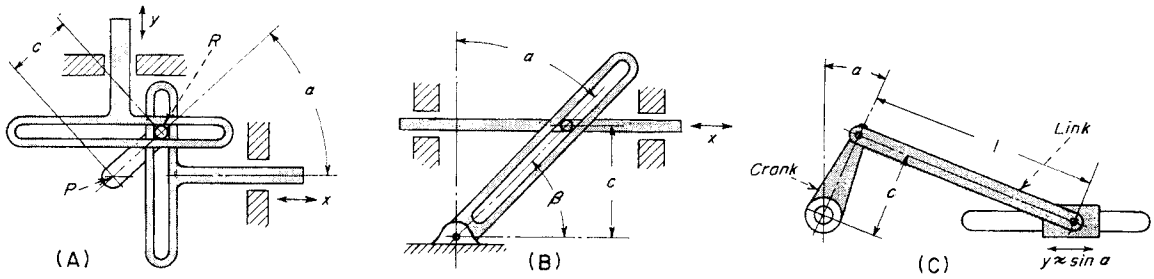



Fig. 4—TRIGONOMETRIC FUNCTIONS. (A) Scotch-yoke mechanism for sine and cosine functions. Crank rotates about fixed point $P$ generating angle $\alpha$ and giving motion to arms: $y = c \sin \alpha$; $x = c \cos \alpha$. (B) Tangent-cotangent mechanism generates: $x = c \tan \alpha$ or $x = c \cot \beta$. (C) Eccentric and follower is easily manufactured but sine and cosine functions are approximate. Maximum error is: $e$ max $= l - \sqrt{l^2 - c^2}$; error is zero at 90 and 270 deg. $l$ is the length of the link and $c$ is the length of the crank.

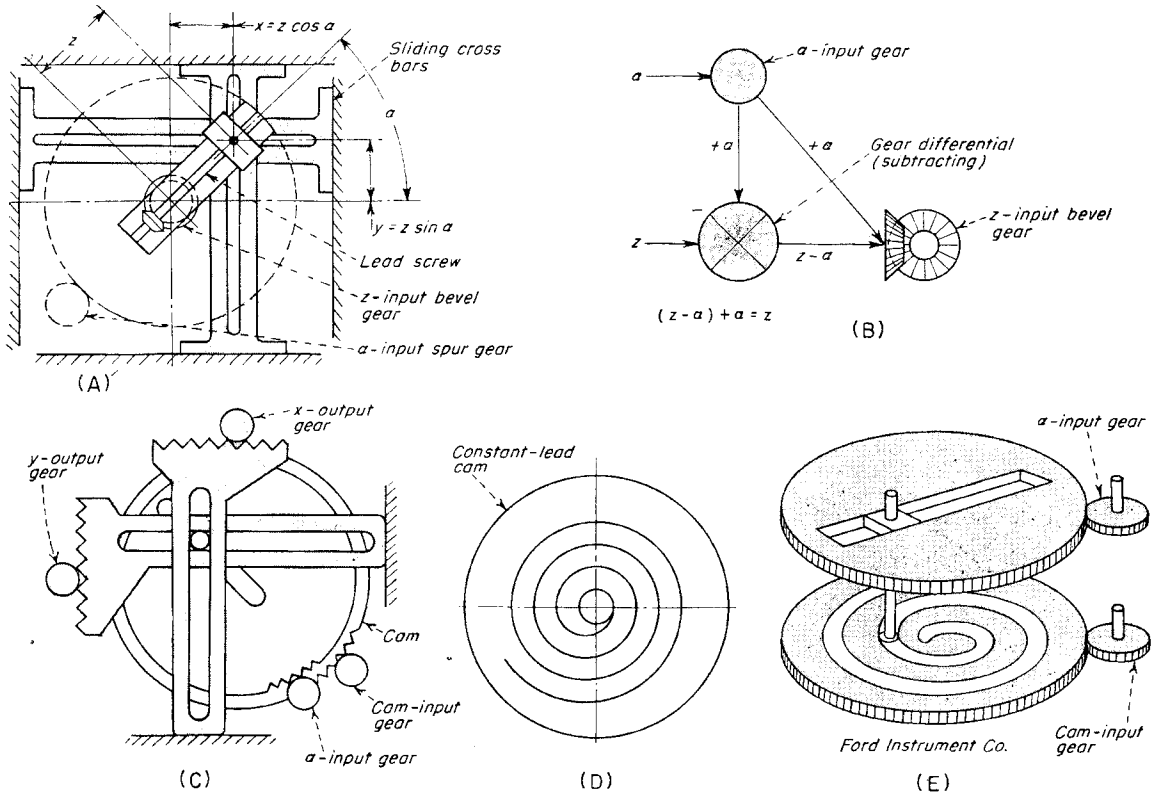



Fig. 5—COMPONENT RESOLVERS for obtaining $x$ and $y$ components of vectors that are continuously changing in both angle and magnitude. Equations are: $x = z \cos \alpha$, $y = z \sin \alpha$ where $z$ is magnitude of vector, and $\alpha$ is vector angle. Mechanisms can also combine components to obtain resultant. Input in (A) are through bevel gears and lead screws for $z$ input, and through spur gears for $\alpha$-input. Compensating gear differential (B) prevents $\alpha$-input from affecting $z$-input. This problem solved in (C) by using constant-lead cam (D) and (E).

# Computing Mechanisms II

**Several typical computing mechanisms for performing the mathematical operations of multiplication, division, differentiation, and integration of variable functions. Analog computing mechanisms for adding and subtracting, for resolving vectors, and for trigonometric functions, are discussed in Part I.**

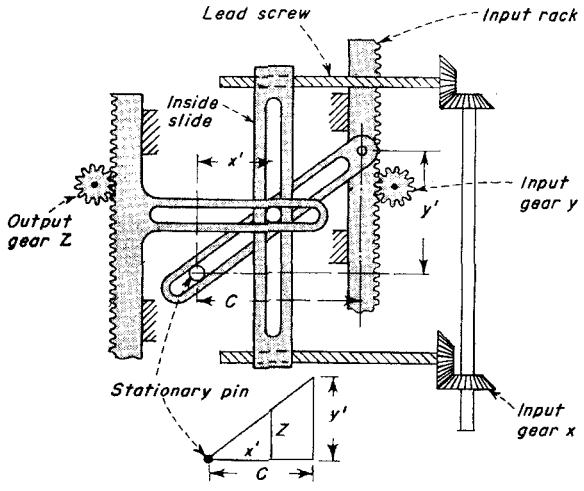



Fig. 1 (A)

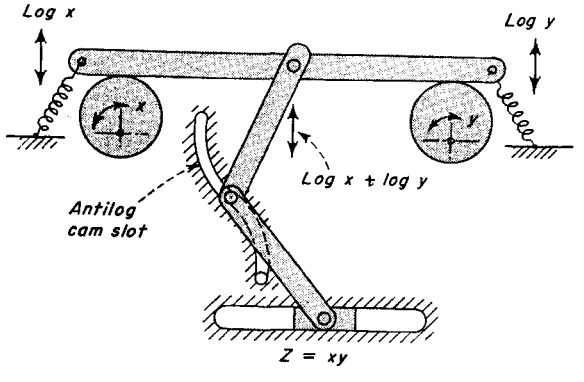



Fig. 1 (B)

**Fig. 1—MULTIPLICATION OF TWO TABLES $x$ and $y$ usually solved by either: (A) Similar triangle method, or (B) logarithmic method. In (A), lengths $x'$ and $y'$ are proportional to rotation of input gears $x$ and $y$. Distance $c$ is constant. By similar triangles: $z/x=y/c$ or $z=xy/c$, where $z$ is vertical displacement of output rack. Mechanism can be modified to accept negative variables. In (B), input variables are fed through logarithmic cams giving linear displacements of log $x$ and log $y$. Functions are then added by a differential link giving $z=\log x + \log y=\log xy$ (neglecting scale factors). Result is fed through antilog cam; motion of follower represents $z=xy$.**

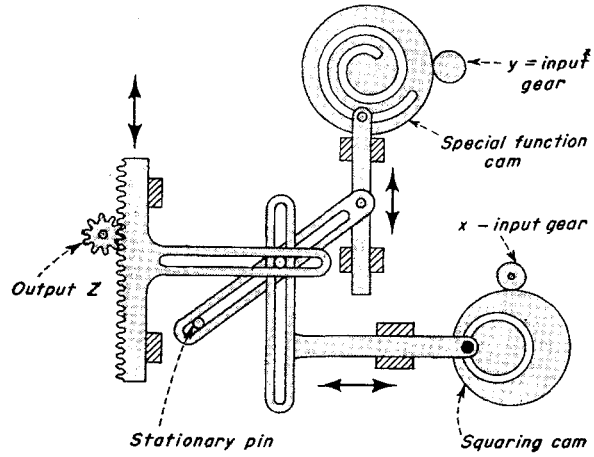



Fig. 2 (A)

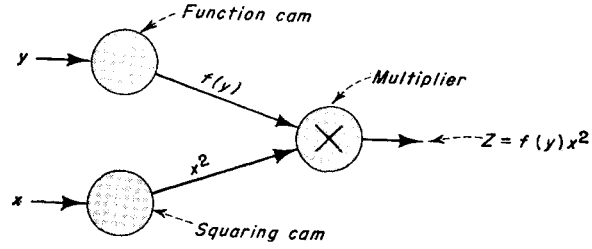



Fig. 2 (B)

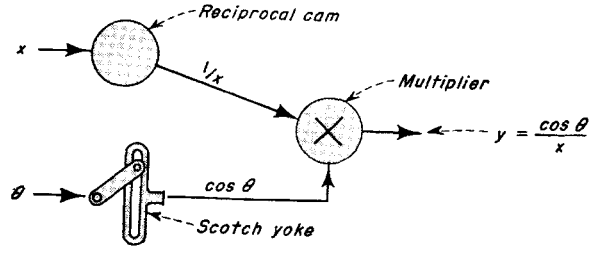



Fig. 2 (C)

**Fig. 2—MULTIPLICATION OF COMPLEX FUNCTIONS can be accomplished by substituting cams in place of input slides and racks of mechanism in Fig. 1. Principle of similar triangles still applies. Mechanism in (A) solves the equation: $z=f(y)\,x^2$. Schematic is shown in (B). Division of two variables can be done by feeding one of the variables through a reciprocal cam and then multiplying it by the other. Schematic in (C) shows solution of $y=\cos\theta/x$.**

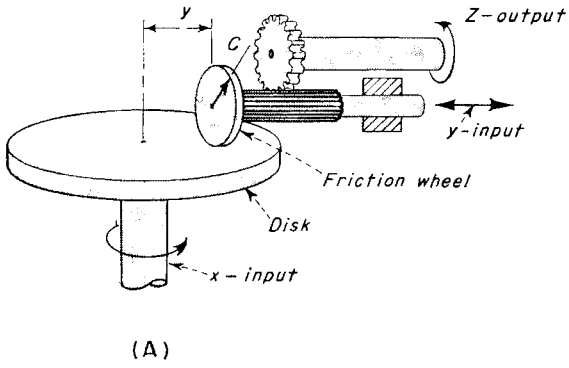



Fig. 3—INTEGRATORS are basically variable speed drives. Disk in (A) is rotated by x-input which, in turn, rotates the friction wheel. Output is through gear driven by spline on shaft of friction wheel. Input y varies the distance of friction wheel from center of disk. For a wheel with radius c, rotation of disk through infinitesmal turn dx causes corresponding turn dz equal to: dz=(1/c) y dx. For definite x revolutions, total z revolutions will be equal to the integral of (1/c) y dx, where y varies as called for by the problem. Ball integrator in (B), gives pure rolling in all directions.

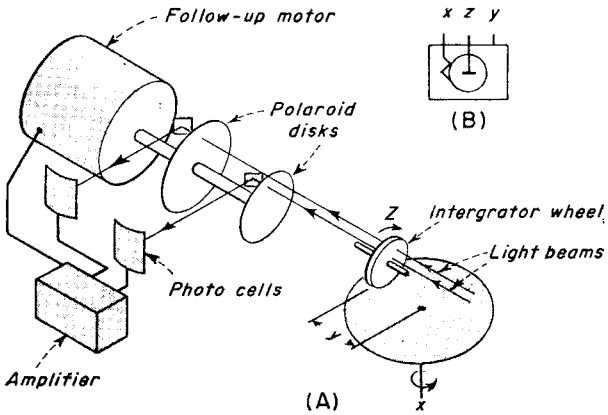



Fig. 4—FOLLOW-UP MOTOR avoids slippage between wheel and disk of integrator in Fig. 3 (A). No torque is taken from wheel except to overcome friction in bearings. Web of integrator wheel is made of polaroid. Light beams generate current to amplifier which controls follow-up motor. Symbol at upper right corner is schematic representation of integrator. For more information see, "Mechanism," by Joseph S. Beggs, McGraw-Hill Book Co., N. Y., 1955.

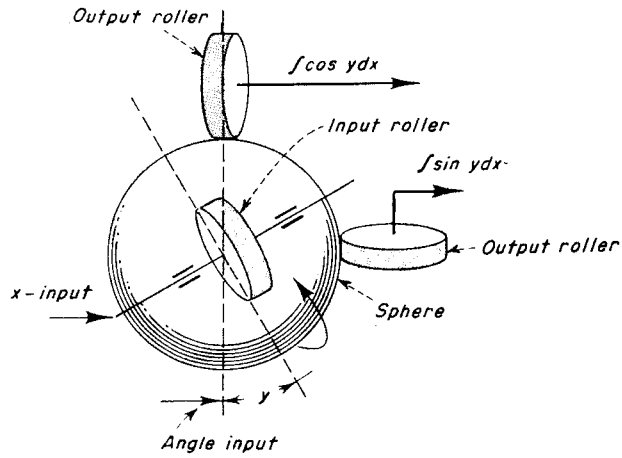



Fig. 5—COMPONENT INTEGRATOR uses three disks to obtain $x$ and $y$ components of a differential equation. Input roller $x$ spins sphere; $y$ input changes angle of roller. Output rollers give integrals of components paralleling $x$ and $y$ axes. Ford Instrument Company.

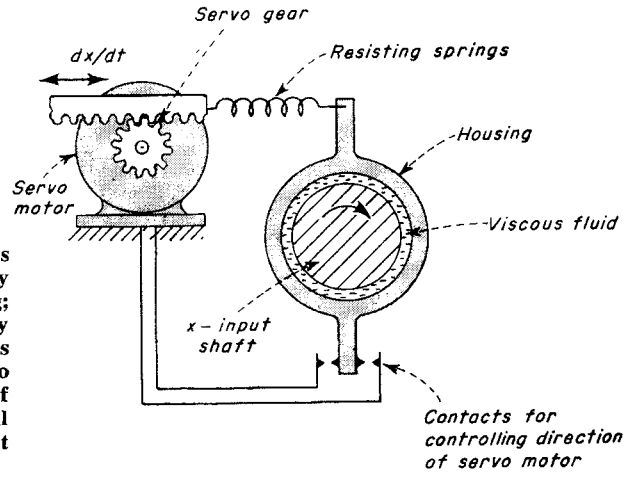


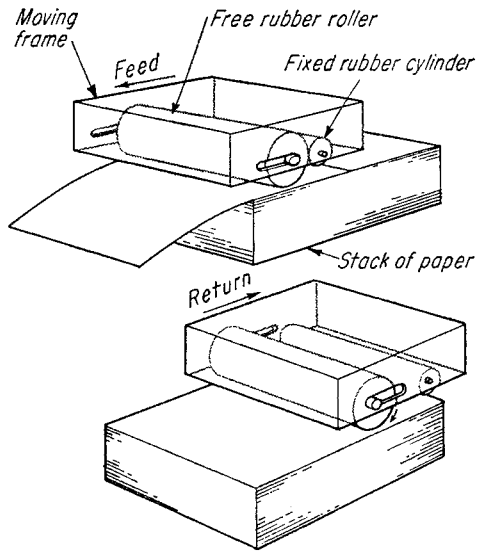

Fig. 6—DIFFERENTIATOR uses principle that viscous drag force in thin layer of fluid is proportional to velocity of rotating x-input shaft. Drag force counteracted by spring; spring length regulated by servo motor controlled by contacts. Change in shaft velocity causes change in viscous torque. Shift in housing closes contacts causing motor to adjust spring length and balance system. Total rotation of servo gear is proportional to dx/dt. From "Mechanical Computing Mechanisms," Reid and Stromback, Product Engineering, October 1949 page 126.

# Eight Paper-Feed Mechanisms

**In a world of reports, forms and memos, a prime need is to keep paper moving. Here are some ways that do the job for single sheets or continuous strips.**

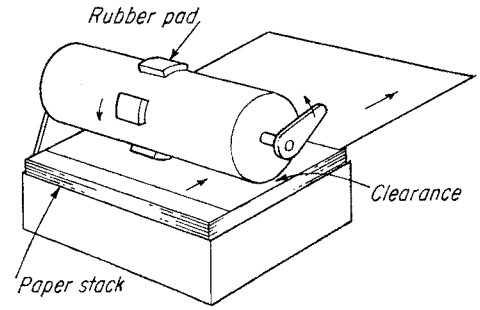
Frank William Wood JR & Vernal Huffines

## SINGLE-SHEET FEEDERS



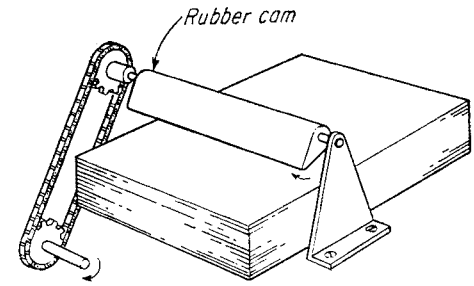

**1**

**FREE ROLLER** rides in slot. During feed it jams against the fixed cylinder and grips top sheet of paper. Return motion of frame transfers free roller to opposite end of slot, where it's free to roll back over paper.




**2**

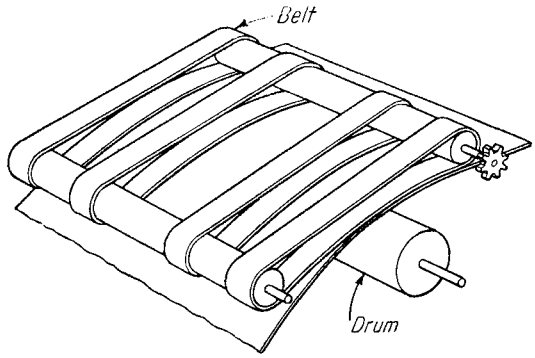
**RUBBER PADS** on rotating cylinder kick out one sheet per pad every revolution. Constant-force spring under paper-holder maintains proper clearance between paper and cylinder. Spacing of pads and the cylinder speed determine feed rate.




**3**

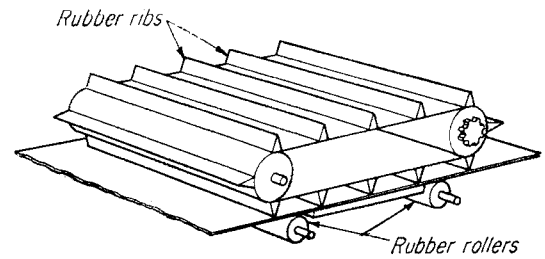
**RUBBER CAM** feeds one sheet each revolution. Constant-force spring under paper table keeps correct clearance. By correct timing, two or more cams in a stack will deliver different sheets in sequence. As with all single-sheet feeders, paper must be smooth enough to slide off pile.

## CONTINUOUS-STRIP FEEDERS



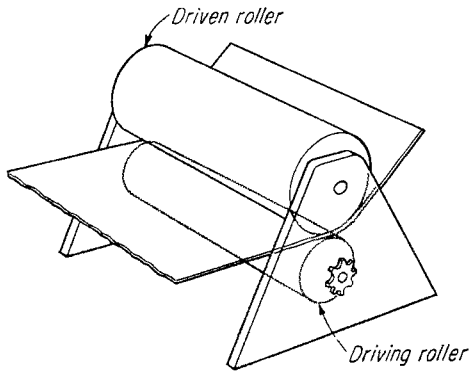

**4**

**BELTS** pressing against drum allow paper to slip and stay in alignment.



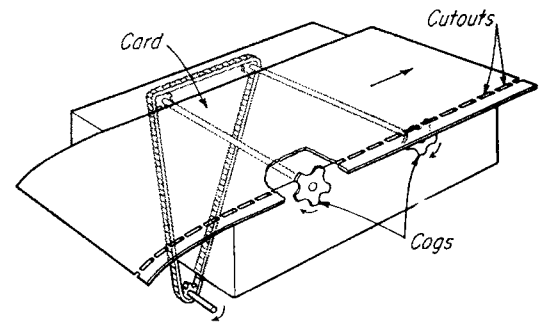

**5**

**RIBBED BELT** is another type of feed that allows paper to slip.



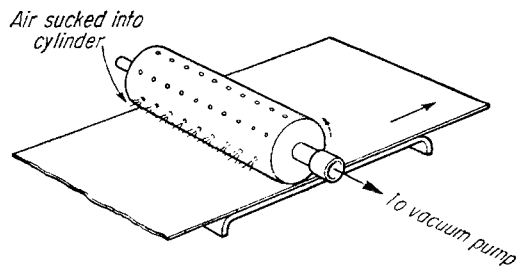

**6**

**FRICTION ROLLERS** are the commonest, cheapest way to feed paper in continuous strips.




**7**

**COGS** fit in perforations to give positive feed with no slipping.



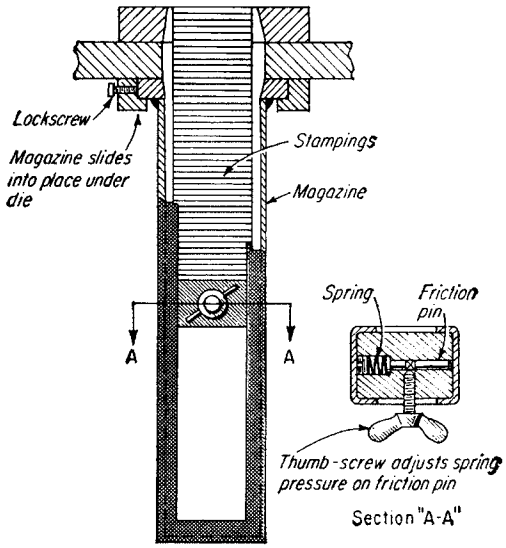

**8**

**VACUUM PUMP** sucks air in through the holes, holding paper against the cylinder. Intermittent operation of vacuum keeps paper from wrapping around cylinder.

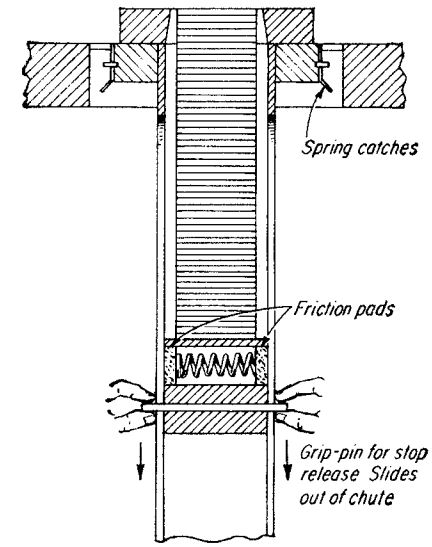
# How to Collect and Stack Die Stamping

**When you're called upon to design a way to collect die stampings as they come from the press, do you shopping here for ideas.**

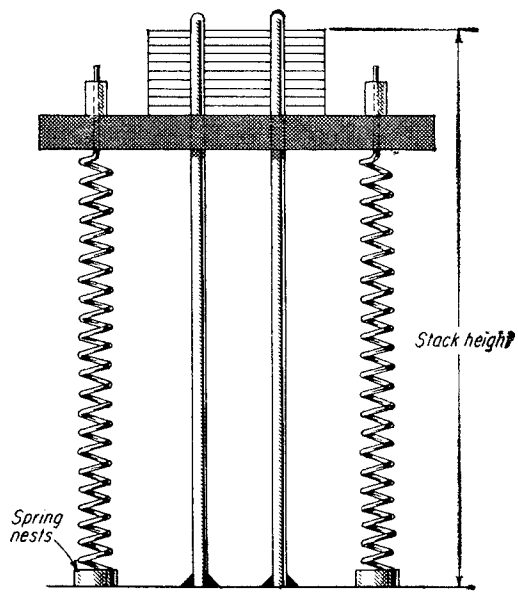
Federico Strasser



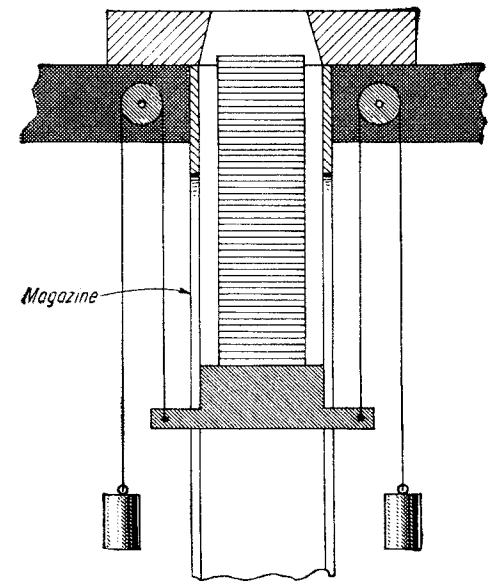

**1.** SLIDING MAGAZINE




**2.** HANGING MAGAZINE

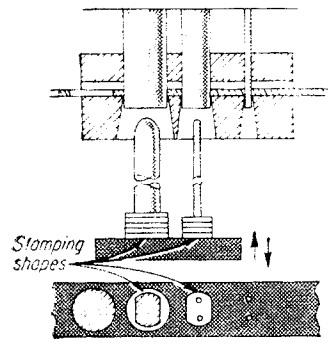



**3.** SPRINGS — stack height is constant when stamping weight divided by spring rate equals stamping thickness

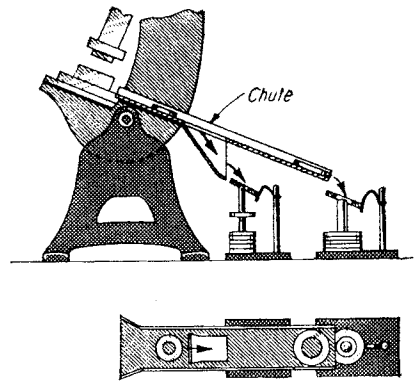



**4.** COUNTERBALANCE METHOD does not need high stacking friction as in first two methods shown above

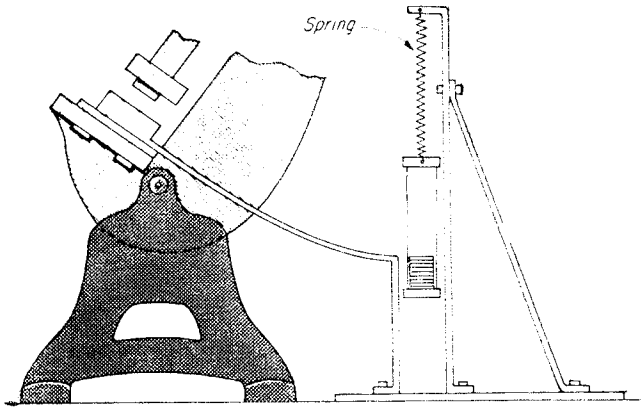
**DIRECT STACKING WITH BACK PRESSURE.** FRICTION DRAGS, SPRINGS, AND COUNTERBALANCE WEIGHTS PROVIDE BACK PRESSURE TO ALLOW SMOOTH STACKING IN FIRST FOUR METHODS



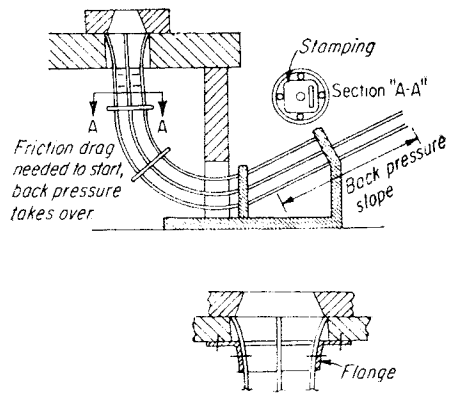

**5.** PEGGED MAGAZINE lets stampings be stacked by gravity, guided by hole in stamping



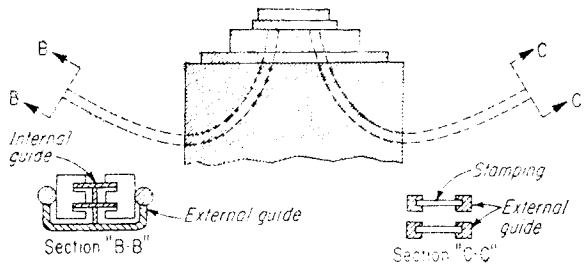

**6.** CHUTED STACKING WITH PEGGED MAGAZINE "sieves" stampings from two-stage die



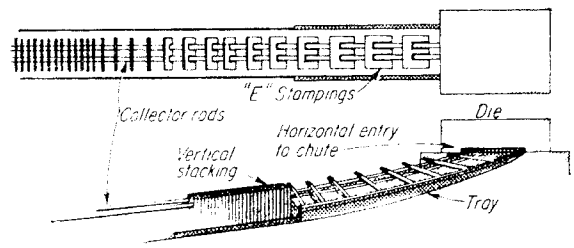

**7.** CHUTED STACKING WITH SPRING-SUSPENDED MAGAZINE




**8.** RODS FORM CHUTE




**9.** SPECIAL SHAPES ARE CHUTED IN CORRECT ORIENTATION




**10.** E-STAMPINGS ARE CHUTED FROM HORIZONTAL TO VERTICAL STACKING ON RODS

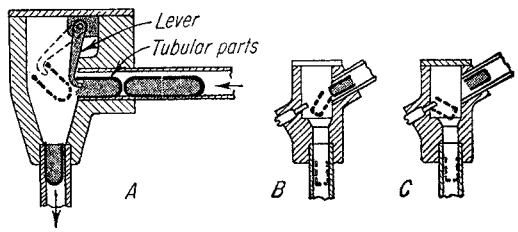
# Sort, Feed, or Weigh Mechanisms

**Sooner or later, you may be faced with the need to design such devices for your plant. These 19 selections are easily modified to suit your product.**

Nicholas P. Chironis

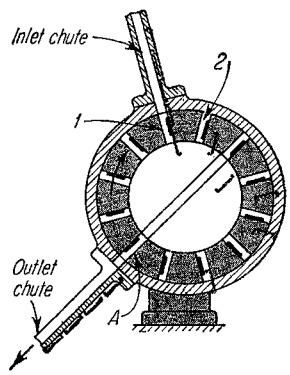
## ORIENTING DEVICES

### Orienting short, tubular parts



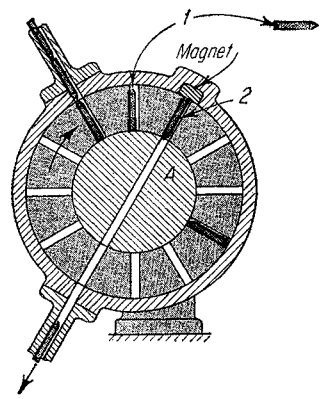

Here's a common problem: Parts come in either open-end or closed-end first; you need a device which will orient all the parts so they feed out facing the same way. In (A), when a part comes in open-end first, it is pivoted by the swinging lever so that the open end is up. When it comes in closed-end first, the part brushes away the lever to keel over head-first. Fig B and C show a simpler arrangement with pin in place of lever.

### Orienting dish-like parts



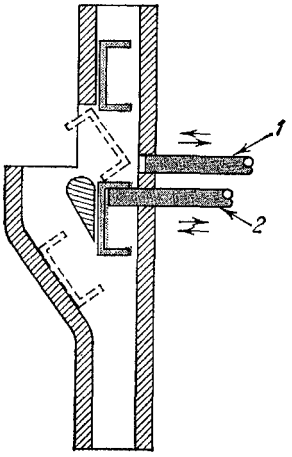

Part with open-end facing to the right (part 1) falls on to a matching projection as the indexing wheel begins to rotate clockwise. The projection retains the part for 230 deg to point *A* where it falls away from the projection to slide down the outlet chute, open-end up. An incoming part facing the other way (2) is not retained by the projection, hence slides *through* the indexing wheel so that it, too, passes through the outlet with the open-end up.

### Orienting pointed-end parts



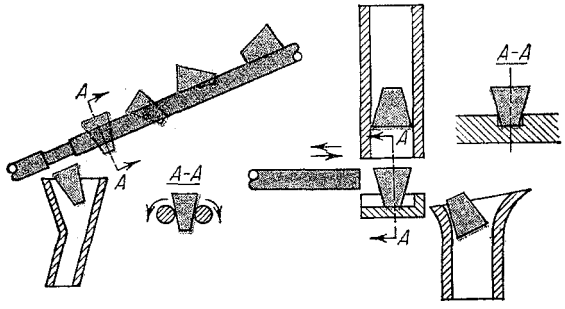

Main principle here is that the built-in magnet cannot hold on to a part as it passes by if the part has its pointed end facing the magnet. Such a correctly oriented part (part 1) will fall through the chute as the wheel indexes to a stop. An incorrectly oriented part (part 2) is briefly held by the magnet until the indexing wheel continues on past the magnet position. The wheel and the core with the slot must be made of nonmagnetic material.

**Orienting U-shaped parts**




Key to this device is two pins which reciprocate one after another in the horizontal direction. The parts come down the chute with the bottom of the U facing either to the right or left. All pieces first strike and rest on pin 2. Pin 1 now moves into the passage way, and if the bottom of the U is facing to the right, the pin would kick over the part as shown by the dotted lines. If on the other hand the bottom of the U had been to the left, motion of pin 1 would have no effect, and as pin 2 withdrew to the right, the part would be allowed to pass down through the main chute.

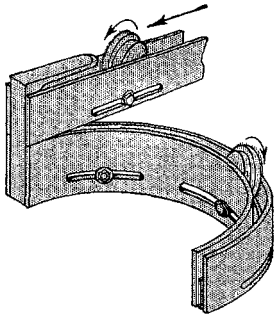
***Orienting cone-shaped parts***




Regardless of which end of the cone faces forward as the cones slide down the cylindrical rods, the fact that both rods rotate in opposite directions causes the cones to assume the position shown in section A-A (left). When the cones reach the thinned-down section of the rods, they fall down into the chute as illustrated.

In the second method of orienting cone-shaped parts (right), if the part comes down small end first, it will fit into the recess. The reciprocating rod, moving to the right, will then kick the cone over into the exit chute. But if the cone comes down with its large end first, it sits on top of the plate (instead of inside the recess), and the rod merely pushes it into the chute without turning it over.

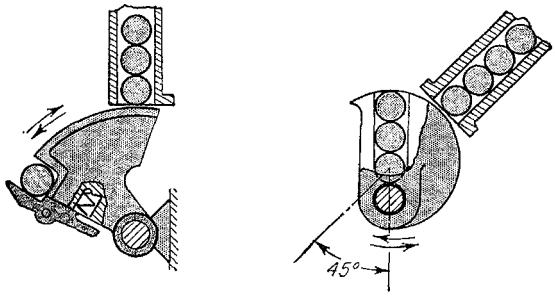
**Orienting stepped-disk parts**



Parts rolling down the top rail to the left drop to the next rail which has a circular segment. The parts, therefore, continue to roll on in the original direction but their faces have now been rotated 180 deg. The idea of dropping one level may seem over simplified, but it avoids the use of camming devices which is the more common way of accomplishing this job.

# SIMPLE FEEDING DEVICES

**Feeding a fixed number of parts**

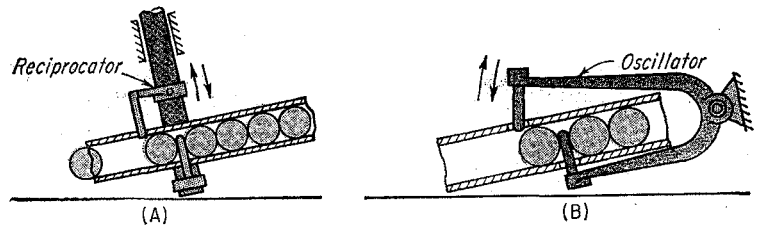



The oscillating sector picks up the desired number of parts, left diagram, and feeds them by pivoting the required number of degrees. The device for oscillating the sector must be able to produce dwells at both ends of the stroke to allow sufficient time for the parts to fall in and out of the sector.

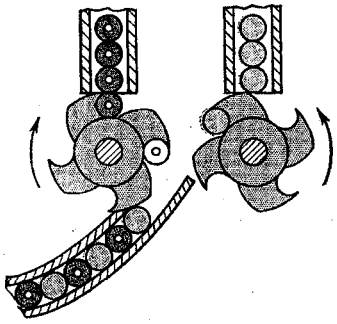
The circular parts feed down the chute by gravity, and are separated by the reciprocating rod. The parts first roll to station 3 during the downward stroke of the reciprocator, then to station 1 during the upward stroke; hence the time span between parts is almost equivalent to the time it takes for the reciprocator to make one complete oscillation.

The device in (B) is similar to the one in (A), except that the reciprocator is replaced by an oscillating member.

**One-by-one separating device**

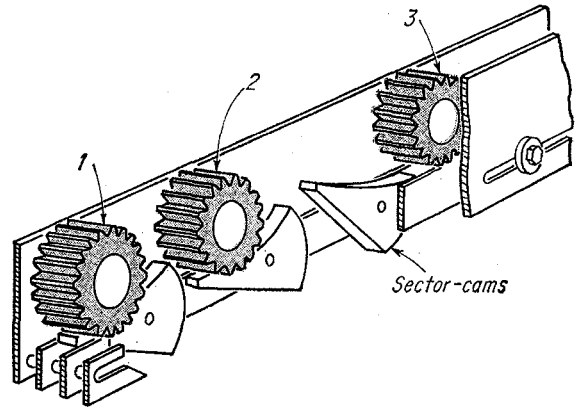



**Mixing different parts together**



Two counter rotating wheels form a simple device for alternating the feed of two different workpieces.

**Pausing until actuated**



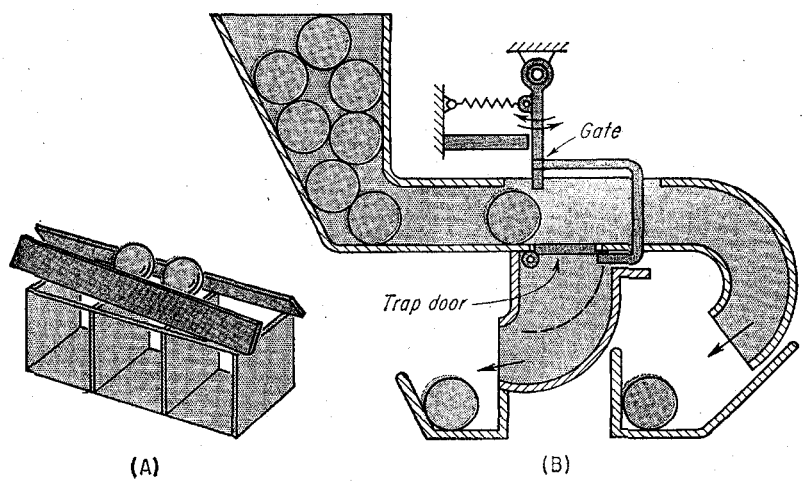

Each gear in this device is held up by a pivotable cam sector until the gear ahead of it moves forward. Thus, gear 3, rolling down the chute, kicks down its sector cam but is held up by the previous cam. When gear 1 is picked off (either manually, or mechanically), its sector cam pivots clockwise because of its own weight. This permits gear 2 to move into the place of gear 1—and frees cam 2 to pivot clockwise. Thus, all gears in the row move forward one station.

# SORTING DEVICES

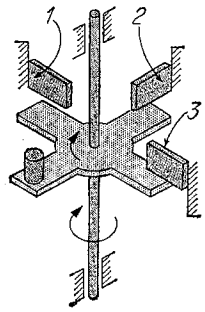
In the simple device (A) the balls run down two inclined and slightly divergent rails. The smallest balls, therefore, will fall into the left chamber, the medium-size ones into the middle-size chamber, and the largest ones into the right chamber.

In the more complicated arrangement (B) the balls come down the hopper and must pass a gate which also acts as a latch for the trapdoor. The proper-size balls pass through without touching (actuating) the gate. Larger balls, however, brush against the gate which releases the catch on the bottom of the trapdoor, and fall through into the special trough for the rejects.

**Sorting balls according to size**



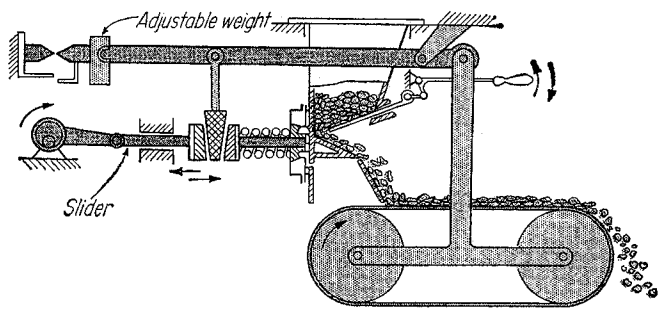
**Sorting according to height**




This is a simple device in which an assembly worker can place the workpiece on a slowly rotating cross-platform. Bars 1, 2, and 3 have been set at decreasing heights beginning with the highest bar (bar 1), and the lowest bar (bar 3). The workpiece is therefore knocked off the platform at either station 1, 2, or 3 depending on its height.

# WEIGHT-REGULATING ARRANGEMENTS

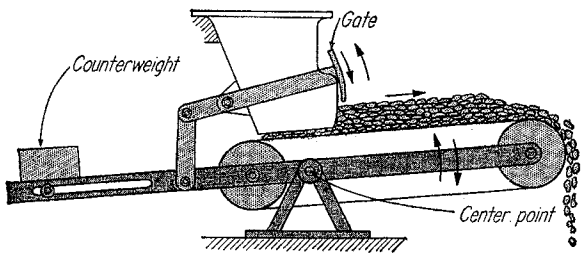
**By varying the vibration amplitudes**




The material in the hopper is fed to a conveyor by means of vibration actuated by the reciprocating slider. The pulsating force of the slider passes through the rubber wedge and on to the actuating rod. The amplitude of this force can be varied by moving the wedge up or down. This is done automatically by making the conveyor pivot around a central point. As the conveyor becomes overloaded, it pivots clockwise to raise the wedge which reduces the amplitudes—and the feed rate of the material.

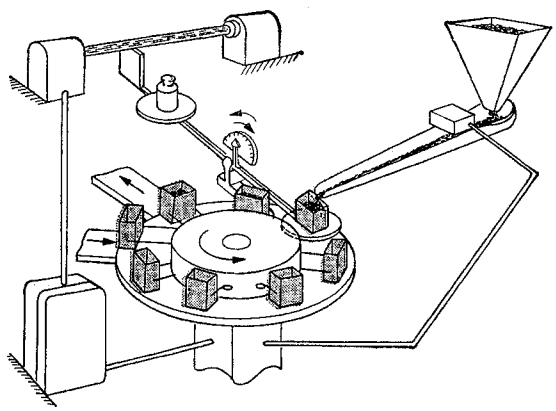
Further adjustments in feed rate can be made by shifting the adjustable weight or by changing the speed of the conveyor belt.

**By linkage arrangement**




The loose material comes down the hopper and is fed to the right by the conveyor system which can pivot about the center point. The frame of the conveyor system also actuates the hopper gate, so that if the material on the belt exceeds the required amount the conveyor pivots clockwise and closes the gate. The position of the counterweight on a frame determines the feed rate of the system.

**By electric-eye and balancer**



The indexing table automatically comes to a stop at the feed station. As the material drops into the container, its weight pivots the screen upward to cut off the operation of the photocell relay. This in turn shuts the feed gate. Reactuation of the indexing table can be automatic after a time interval, or by the cutoff phase of the electric eye.

**EDITOR'S NOTE:** The above material is based on the book *"Mechanism,"* by C. H. Kohevnikov; published in Russian, Moscow, 1965.

# Spring Motors & Typical Associated Mechanisms

MANY applications of spring motors in clocks, phonographs, motion picture cameras, rotating barber poles, game machines and other mechanisms offer practical ideas for adaptation to any mechanism in which operation for an appreciable length of time is desirable. While spring motors are usually limited to comparatively small power applications where other sources of power are unavailable or impracticable, they may also be useful for intermittent operation requiring

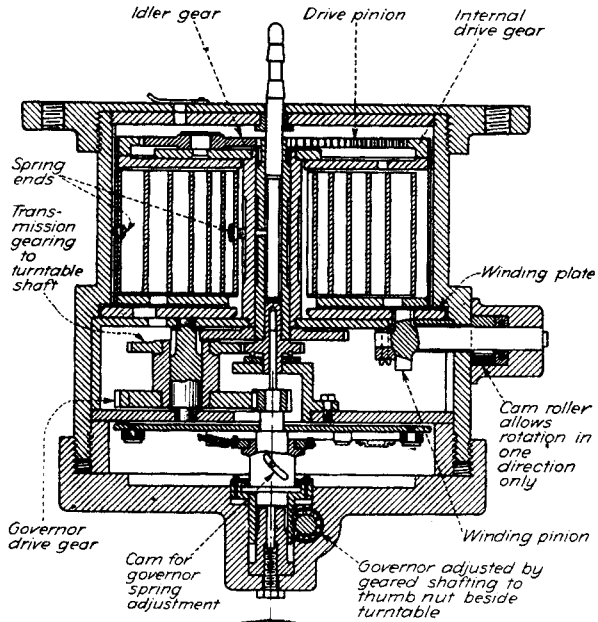
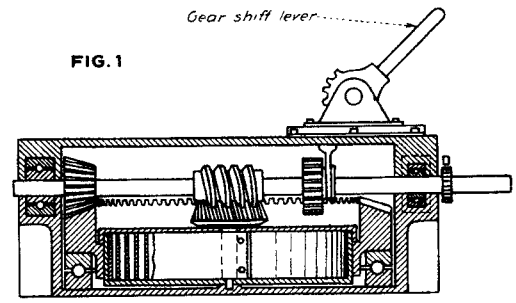

Idler gear
Drive pinion
Internal drive gear
Spring ends
Trans-mission gearing to turntable shaft
Winding plate
Cam roller allows rotation in one direction only
Governor drive gear
Winding pinion
Cam for governor spring adjustment
Governor adjusted by geared shafting to thumb nut beside turntable

FIG. 1



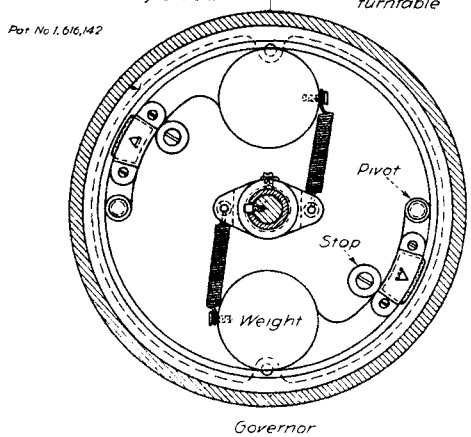
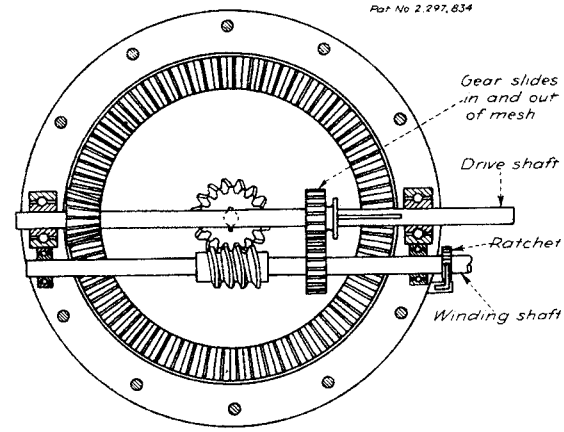




FIG. 2



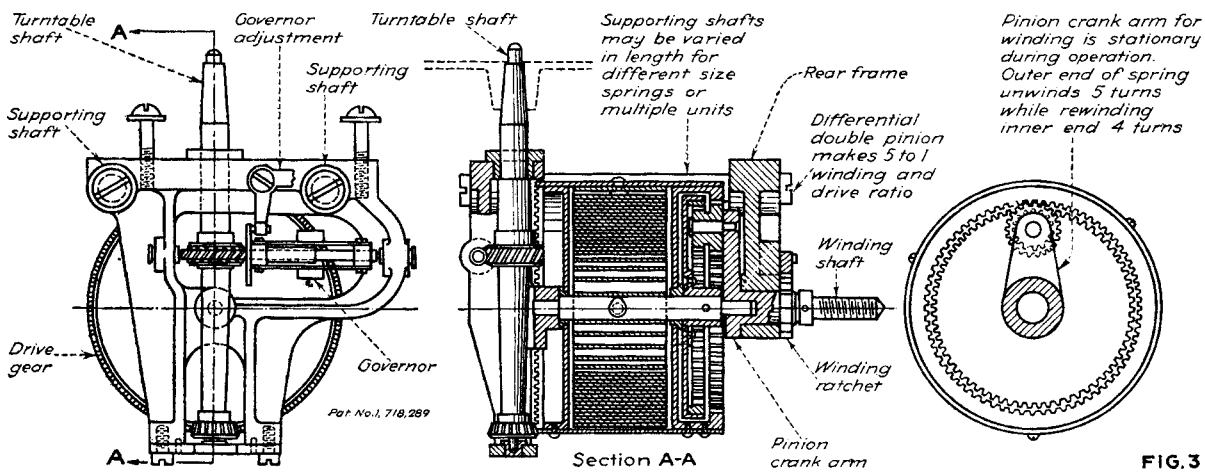


Turntable shaft
A
Governor adjustment
Supporting shaft
Supporting shaft
Drive gear
Governor
Pat No.1, 718,289
A
Turntable shaft
Supporting shafts may be varied in length for different size springs or multiple units
Rear frame
Differential double pinion makes 5 to 1 winding and drive ratio
Winding shaft
Winding ratchet
Pinion crank arm
Section A-A
Pinion crank arm for winding is stationary during operation. Outer end of spring unwinds 5 turns while rewinding inner end 4 turns

FIG. 3

comparatively high torque or high speed, using a low power electric motor or other means for building up energy.

The accompanying patented spring motor designs show various methods of transmission and control of spring motor power. Flat-coil springs, confined in drums, are most widely used because they are compact, produce torque directly, and permit long angular displacement. Gear trains and feed-back mechanisms hold down excess power so that it can be applied for a longer time, and governors are commonly used to regulate speed.

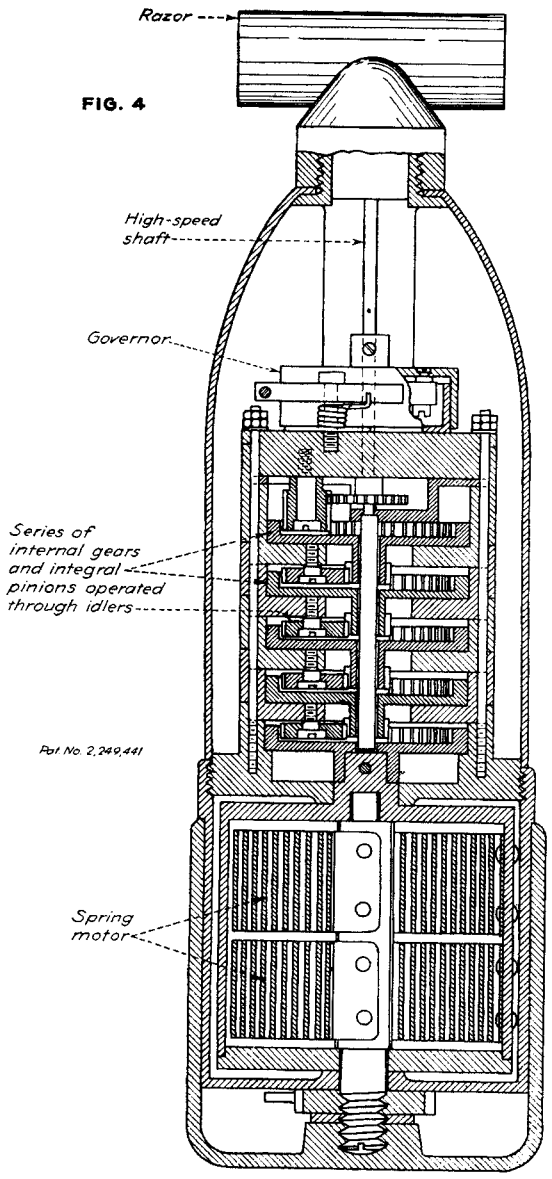
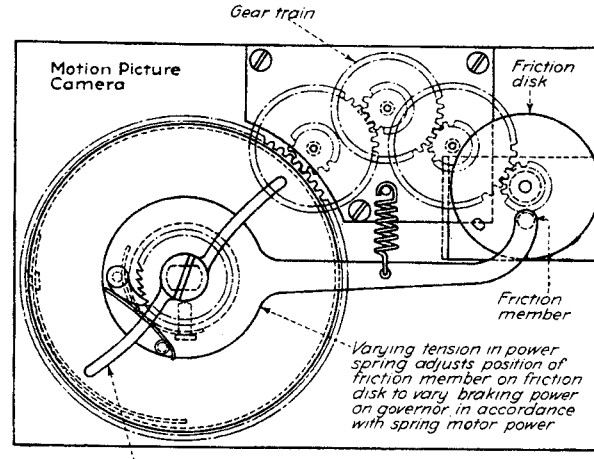



FIG. 4


Gear train
Motion Picture Camera
Friction disk
Friction member
Varying tension in power spring adjusts position of friction member on friction disk to vary braking power on governor in accordance with spring motor power
Pat No. 2,095,670

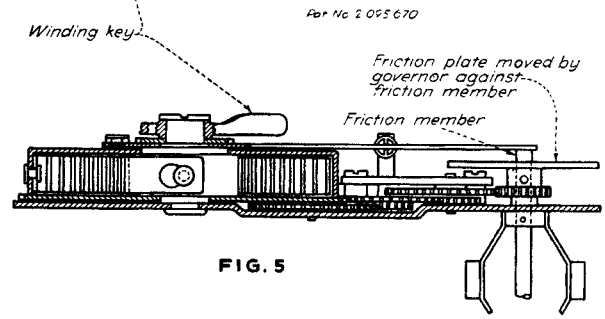



FIG. 5

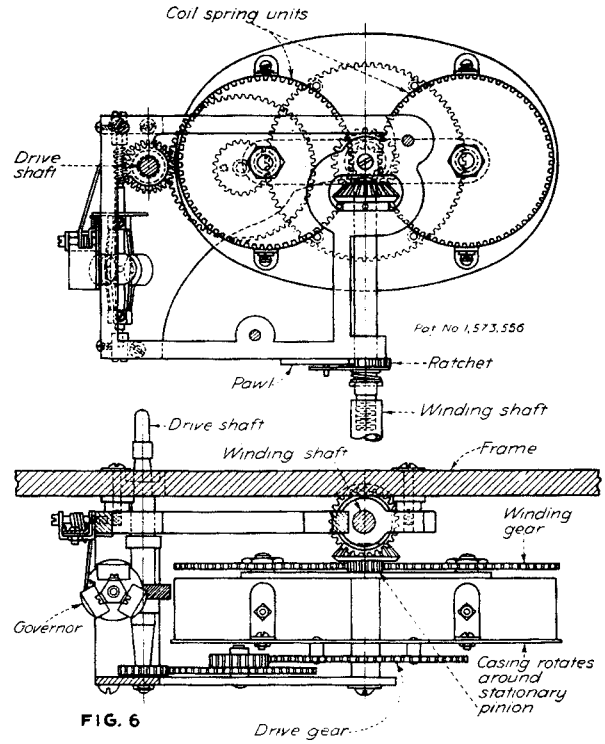


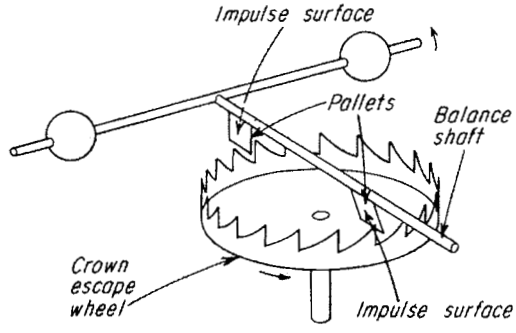

FIG. 6

# 6 Escapement Mechanisms

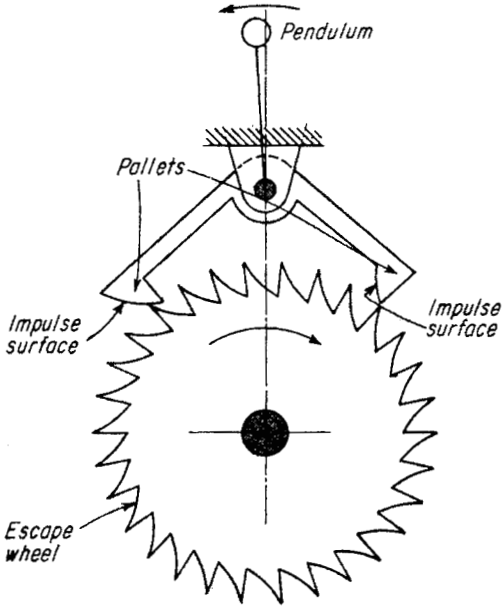
Federico Strasser

Escapements, a familiar part of watches and clocks, can also be put to work in various control devices, and for the same purpose—to interrupt the motion of a gear train at regular intervals.

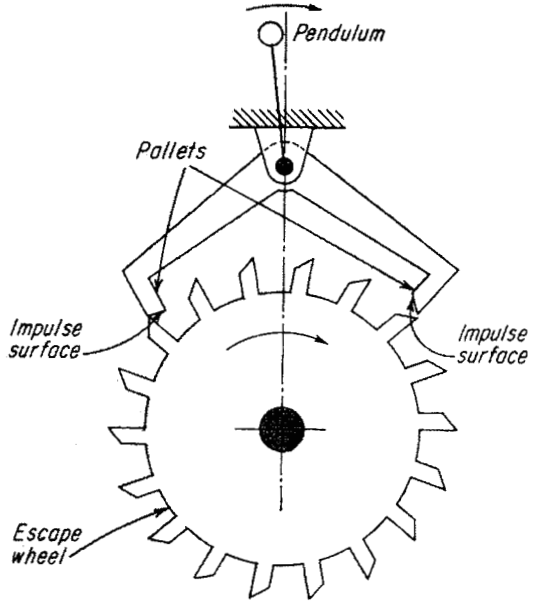
Each escapement shown here contains an energy absorber. Whether balance wheel or pendulum, it allows the gear train to advance one tooth at a time, locking the wheel momentarily after each half-cycle. Energy from the escape wheel is transferred to the escapement by the advancing teeth, which push against the locking surfaces (pallets).



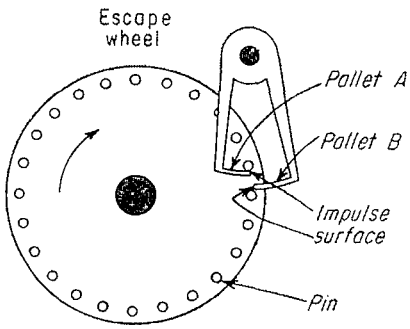

**1 VERGE.** Pallets are alternately lifted and cleared by diametrically opposed teeth on escape wheel, thus rocking balance shaft. One pallet or the other is always in contact with a tooth of the escape wheel.



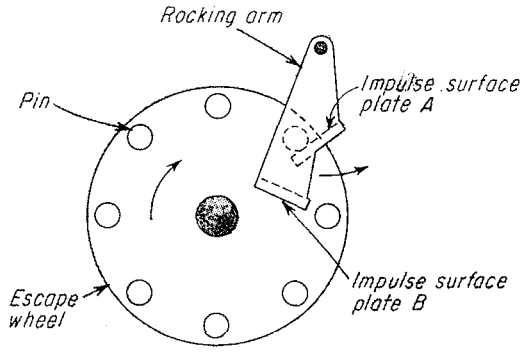

**2 ANCHOR—RECOIL.** As pendulum swings to one side, pallet engages tooth of escape wheel. When wheel is gripped, it recoils slightly. Pendulum starts in opposite direction by push of wheel transmitted through pallet. As pendulum swings to other side, the other pallet engages escape-wheel tooth, recoils, and repeats cycle.



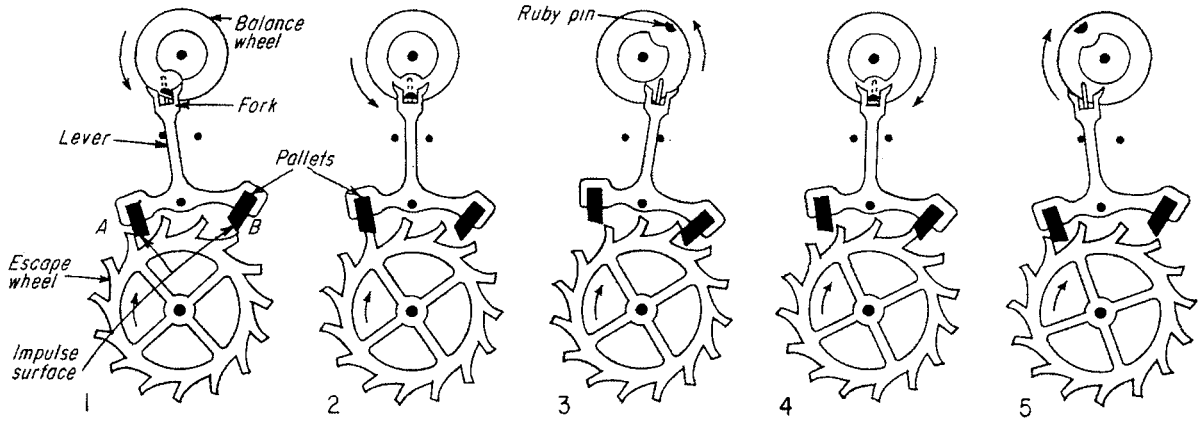

**3 ANCHOR—DEAD BEAT.** For greater accuracy, recoil escapement is modified by changing angle of impulse surfaces of pallets and teeth so that recoil is eliminated. The teeth of escape wheel fall "dead" upon pallets; as pendulum completes its swing, escape wheel does not turn backward.

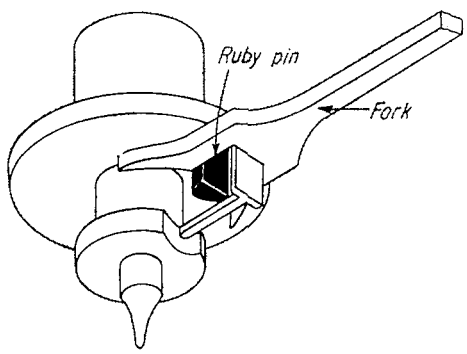



**4 PIN WHEEL.** Pallet *A* is about to unlock escape wheel. Pin will give a push to pallet *A* while sliding down impulse surface. As pendulum continues to swing, pallet *B* will stop pin, thus locking escape wheel. After pendulum reverses direction and swings toward dead center, pallet *B* releases wheel and receives an impulse from pin.




**5 LANTERN WHEEL.** Escape wheel is stopped when pin hits plate *A* of pallet, attached to rocking arm. As arm swings to right, pin pushes against impulse surface of plate until pin is clear. Escape wheel then turns freely until pin is stopped by plate *B* of pallet. Arm starts back toward left, receiving impulse from pin until latter clears plate.



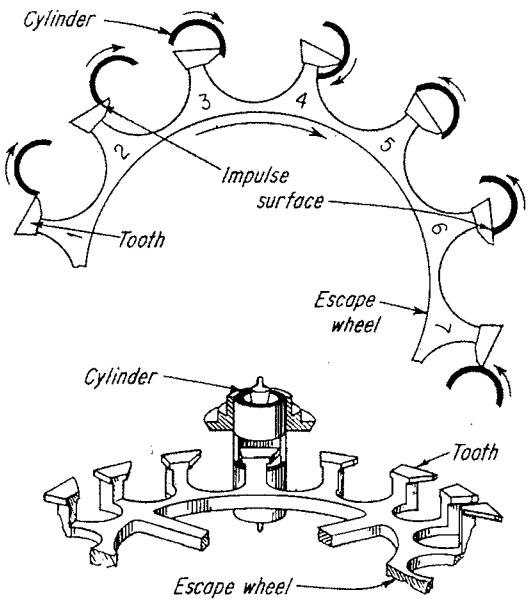




**6 MUDGE'S LEVER.** As balance wheel swings counterclockwise (1), the ruby pin enters fork of lever and pushes it to the right. The lever releases tooth of escape wheel, which starts turning and gives a tiny push (2) to impulse surface of pallet A. This is transmitted to balance wheel through fork and ruby pin. Balance wheel continues rotating counterclockwise (3) while lever swings to right and pallet *B* stops the escape wheel by engaging tooth. When balance wheel reaches its hairspring's limit, it reverses direction and re-enters fork (4). Ruby pin pushes lever to left, and receives impulse when escape wheel starts to move. Balance wheel continues clockwise until it reaches opposite limit of its hairspring. Lever continues moving to left until it again stops escape wheel (5) and awaits return of ruby pin. Another view of Mudge's lever is given in (6). This escapement is very precise because of the small portion of each half-cycle that escape wheel and balance wheel are in contact. Friction drag is at a minimum.

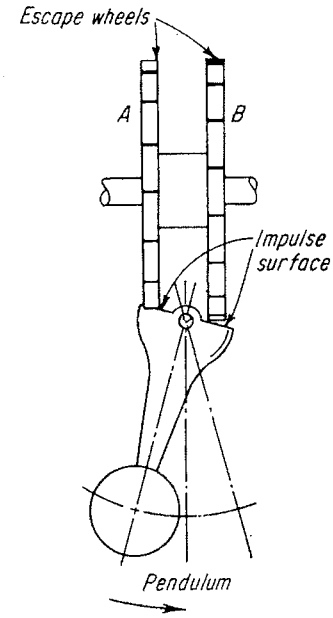
# 7 More Escapement

**Another selection of these ingenious mechanical components for use in control systems.**

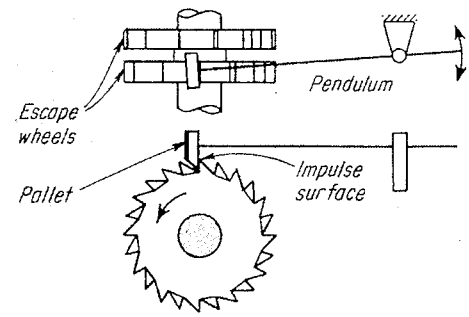
Federico Strasser



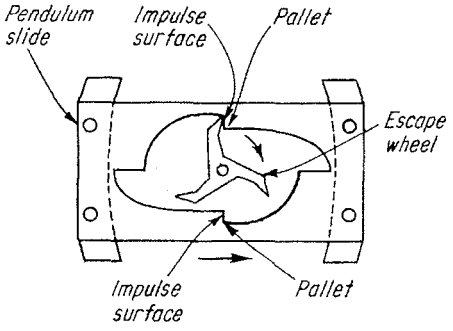

**1 CYLINDER.** Tooth of escape wheel and cylinder (balance-wheel shaft) are shown in seven positions. Tooth has been locked (1) and is about to enter cycle. Tooth is providing impulse (2) by sliding action along cylinder lip. Escape wheel is locked while cylinder rotates clockwise (3) to its limit (4), then starts back (5). As tooth starts forward, impulse is again imparted (6) to cylinder lip. Second tooth is locked (7) during period of backswing.



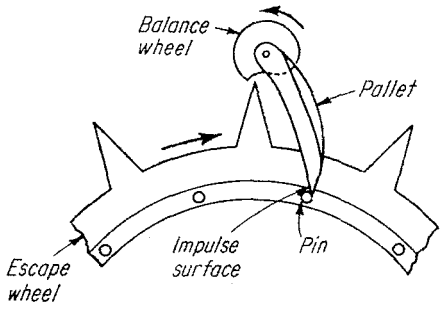

**2 DOUBLE RATCHET.** Two escape wheels are mounted on a common axis and pinned together so teeth are aligned with a half-pitch distance between them. Drawing shows pendulum in its extreme left position, with its impulse surface locking a tooth of the escape wheel *A*. As pendulum starts to right, escape wheels start to turn, imparting a push to the pendulum. As it approaches its extreme right position, pendulum again stops the escape wheels, by engaging a tooth on wheel *B*, then receives a second impulse as it starts toward the left.



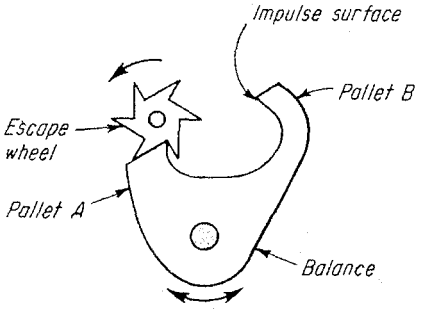

**3 HIGH-SPEED DOUBLE RATCHET.** Same principle as in 2, but used with a small-mass pendulum. Speeds to 50 beats per second can be obtained.



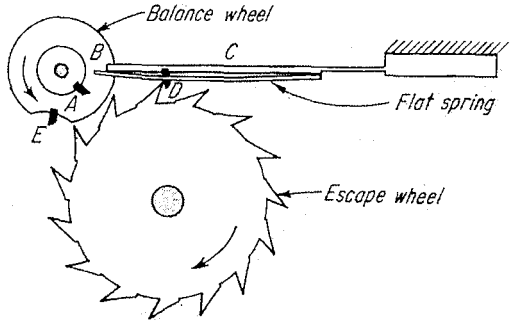

**4 THREE-LEGGED.** The three teeth of escape wheel work alternately upon top and bottom pallets. Top pallet is shown being driven to right. When tooth clears pallet, escape wheel will turn until tooth engages bottom pallet. When pendulum slide starts back toward left, bottom pallet will receive push from escape wheel, until tooth clears pallet.




**5 DUPLEX.** Single pallet of balance wheel is shown receiving impulse from pin of escape wheel. As balance wheel rotates counterclockwise, tooth of escape wheel clears notch, permitting escape wheel to turn clockwise. Next tooth of escape wheel is stopped by balance wheel until balance wheel reaches limit, reverses direction, and pallet swings back to position shown. Escapement receives but one impulse per cycle.




**6 STAR WHEEL.** Dead-beat escapement in which pallets alternately act upon diametrically opposed teeth. Illustration shows balance in extreme clockwise position with escape wheel locked. Balance starts turning counterclockwise, releasing escape wheel. At other extreme, pallet *B* locks escape wheel for a moment before direction of balance is again reversed and push is imparted.

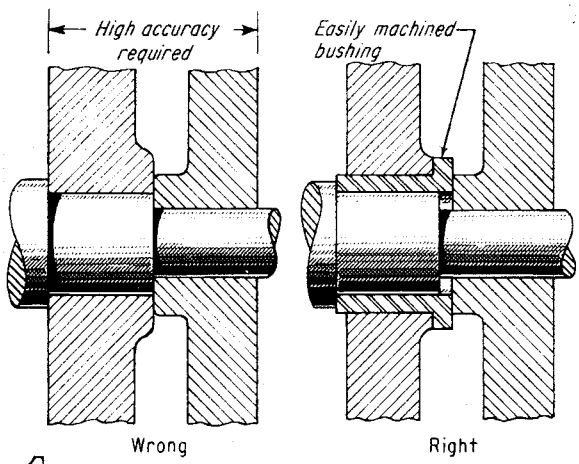



**7 CHRONOMETER.** As balance wheel turns counterclockwise, jewel *A* pushes flat spring *B* and raises bar *C*. Tooth clears jewel *D* and escape wheel turns. A tooth imparts push to jewel *E*. As jewel *A* clears flat spring, jewel *D* returns to position and catches next tooth. On return of balance wheel (clockwise), jewel *A* passes flat spring with no action. Thus, escapement receives a single impulse per cycle.

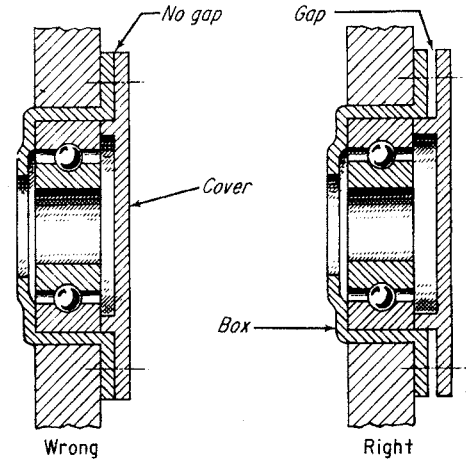
# Design Hints for Mechanical Small Mechanisms

## How to avoid close tolerances, and otherwise improve parts, where accurate assembly is necessary.

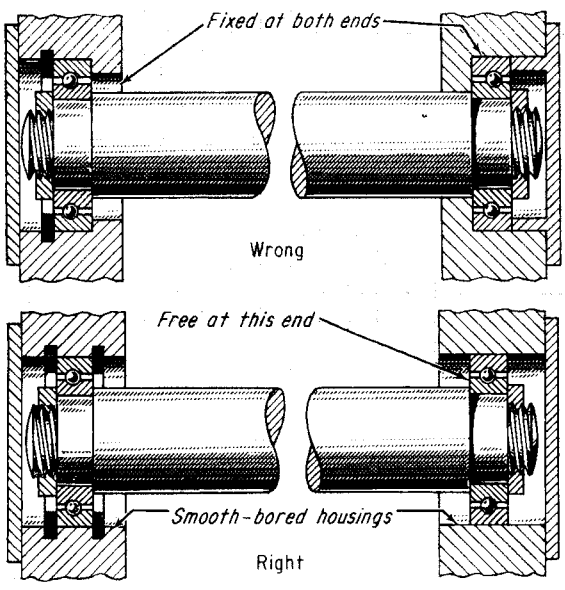
Federico Strasser



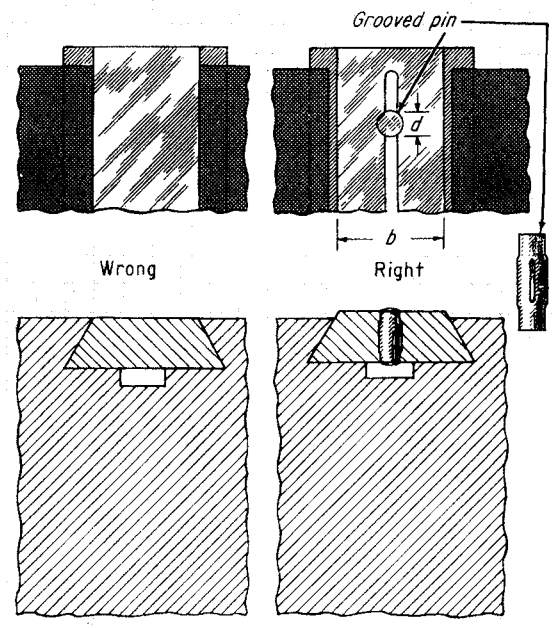

**1** **BUSHING FACE** is much easier to machine than the hub of a large part such as a flywheel or heavy gear.



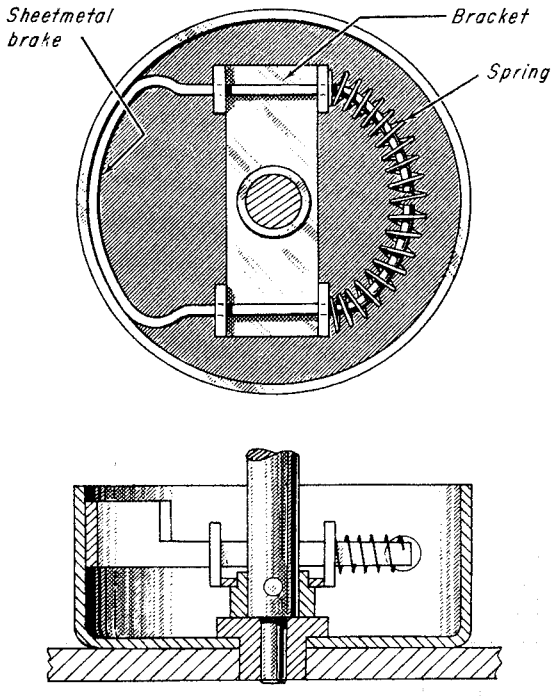

**2** **HOLD-DOWN COVER** for bearing should have a take-up gap between the cover flange and the bearing box.



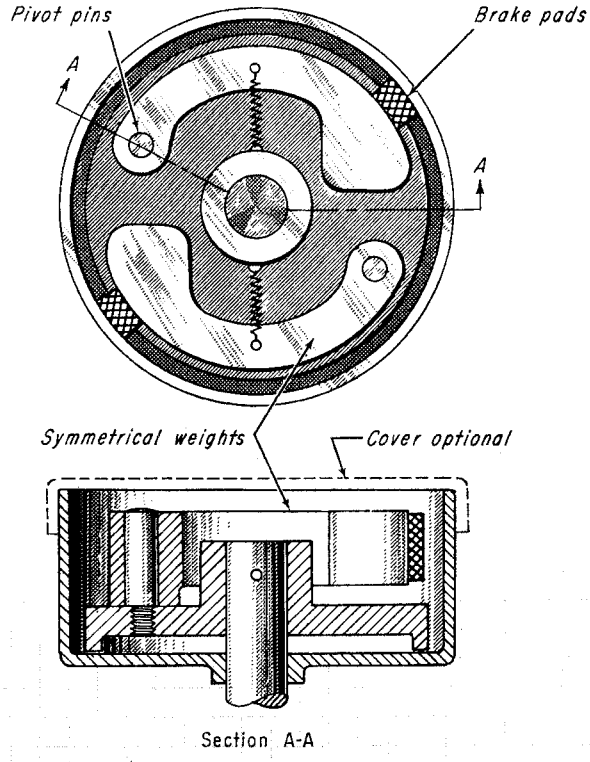

**5** **SMOOTH-BORED HOUSINGS** shown on right let one bearing move when shaft length changes because of expansion.



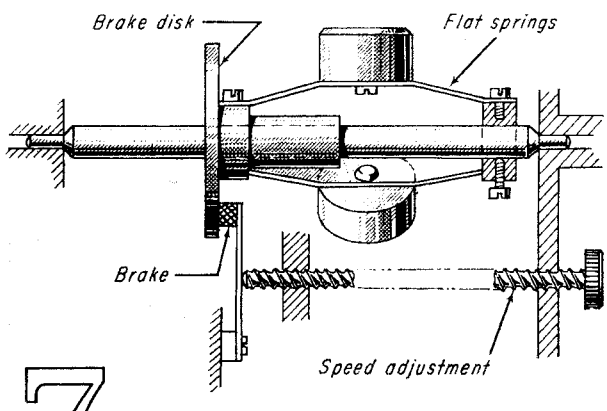

**6** **DOVETAILED** parts that are to be a tight fit are best provided with a slot and grooved pin where $d$ equals $b/3$ to $b/4$.



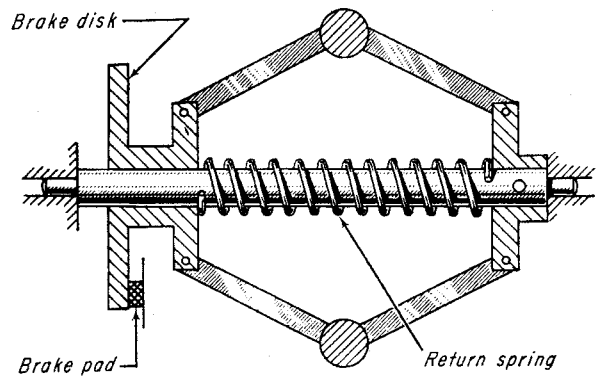

3 **SHEETMETAL BRAKE** provides larger braking area than previous design. Operation is thus more even and cooler.




4 **SYMMETRICAL WEIGHTS** give even braking action when they pivot outward. Entire action can be enclosed.




7 **THREE FLAT SPRINGS** carry weights that provide brake force upon rotation. Device can be provided with adjustment.




8 **TYPICAL GOVERNOR** action of swinging weights is utilized here. As in the previous device, adjustment is optional.